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(Continued on page 29)

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Graham

Gear-Shift Changes Are Spotlighted in 1938 Design Trends

By Austin M. Wolf
Consulting Engineer

THE transmission and its control have received intensive study, and the new offerings now positively indicate the industry's next major development. Other highlights are the Cadillac-16 engine, Cadillac-8 "syncro-flex" flywheel, Buick's turbulator piston, the Chevrolet clutch, the Buick coil-spring suspension, the Packard rear-suspension and axle torque-reaction combination, rayon fabric tires, and the increasing use of the rear-axle transverse stabilizer link. Grilles, sheet-metal work, and body lines collectively have dressed up the exteriors, making new types of styling with fresh distinctive individuality in a few instances.

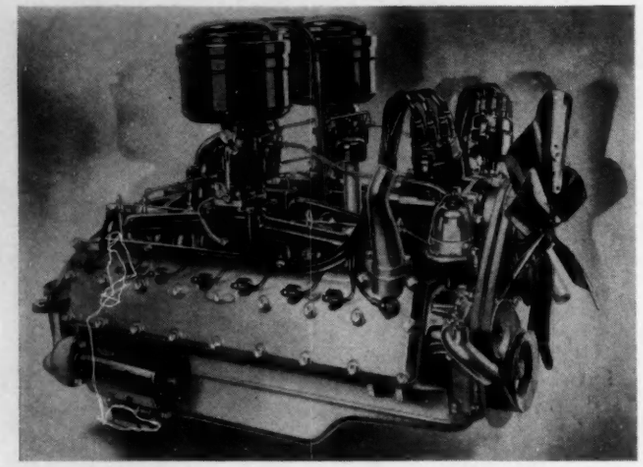
General

Graham has introduced a dashing design and confines itself to two basic models, the Special and the Supercharger. The Chrysler Airflow and the Cadillac "70" and "85" series have been discontinued. The new Cadillac-16 has the same wheelbase as the "75" series and the same Fleetwood body shell. Its distinguishing exterior appearance is obtained by the sheet-metal design. A 3-in. increase in DeSoto's wheelbase brings it up to 119 in. Hupmobile is back in the field with the previous powerplants, the 6 having a 122-in. wheelbase and the 8, a 125-in. wheelbase. The Packard-6 and -8 (the latter replacing the former "120") have an increase of 7 in. in the wheelbase, bringing them to 122 and 127 in., respectively. The seven-passenger 8 has a 148-in. wheelbase. The 1937 Studebaker Dictator is re-christened the Commander and has a companion model, the Studebaker-6, the chassis being identical. The President has dropped the wheelbase from 125 in. to 122 in.

Engines

Engine developments reflect numerous minor changes, but there is one bristling new design, the Cadillac-16. It has a bore and stroke of 3¼ in., a 135-deg. V angle, 431 cu. in. displacement, 7:1 compression ratio, and is lighter even than the former V-12. Inasmuch as the rear portion is located under the toe board, a 39-in. distance from the rear of the radiator core to the dash is possible. A practically continuous flow of power is obtained from the explosion impulses occurring every 45 deg. of crankshaft rotation. With the standard 4.3:1 axle ratio, the piston travel is only 1590 ft. per mile. The cu. ft. per ton-mile are 126. High-gear acceleration of the seven-passenger sedan is in excess of 4.6 ft. per sec. per sec. between 20 and 35 m.p.h. A very low center of gravity is possible with the 135-deg. V angle which permits easy servicing of the intake and exhaust systems and valve gear. Two dual down-draft carburetors are used. Engine and clutch weight is 1050 lb. Block casting of cylinders and upper half of crankcase, nine main bearings, L-head design, siamesed intake, and individual exhaust ports characterize the design. The combustion-chamber is machined all over to avoid compression variation between the cylinders.

With the new combustion-chamber form, Buick has raised the compression ratio on the "40" from 5.7:1 to 6.15:1. The ratio in the large engine has been stepped up from 5.75:1 to



Cadillac 135-Deg. V-16 Engine. Generator is Frictionally Driven and Two Independent Carburetion Systems Are Used. Distributor Construction is Distinctive.

6.35:1. The Nash Ambassador-8 now develops 115 hp. at 3400 r.p.m.; the Ambassador-6 develops 105 hp.; and the Lafayette, 95 hp. as compared to 105, 95, and 90 hp., respectively in 1937. The Packard-6 has increased its bore 1/16 in. bringing it to 3½ in., and the displacement to 245.34 cu. in. The compression ratio has been stepped up from 6.3:1 to 6.52:1 in the 6 and from 6.5:1 to 6.6:1 in the 8. An optional 7.05:1 high-compression head is available for both engines.

The Chrysler Custom Imperial remains unchanged, having the highest compression ratio in the field with the optional 7.45:1 head. The Chrysler Royal has reverted to the 1936 3⅜- by 4½-in. bore and stroke, and is now 241.5 cu. in. in displacement due to the ¼ in. longer stroke. It is interesting to note that the compression ratio is 6.2:1 as against 6.5:1 in 1937 when 93 hp. at 3600 r.p.m. were developed. Now 95 hp. are developed by the new engine. The Chrysler Imperial with 3¼- by 4⅛-in. engine, ⅜ in. longer stroke, now has a displacement of 298.6 cu. in. The 6.2:1 compression ratio supplants last year's 6.7:1 ratio; 110 hp. is developed at 3400 r.p.m., whereas the same output was developed at 3600 r.p.m. last year. Drop in compression ratios is also reflected in the Studebaker President with 6:1, developing 110 hp. at 3600 r.p.m. as compared to last year's 6.5:1 developing 115 hp. at the same speed. The Commander bore has been increased 1/16 in. to 3 5/16 in. The Graham Special and Supercharger engines are identical except for the supercharger, the engines developing 90 hp. at 3600 r.p.m. and 116 hp. at 4000 r.p.m., respectively. The 6.7:1 compression ratio remains unchanged over last year. The low-speed acceleration of the Cadillac "65" has been improved by axle-ratio changes, increasing the cu. ft. per ton-mile to 121, as compared to 116 in 1937. The output of the Series "75" engine has been increased by raising the compression ratio to 6.7:1 based on the use of ethyl fuel. The other Cadillac V-8 engines remain at 6.25:1 compression ratio for use on standard fuel. The change in compression ratio does not affect service stock requirements since only two pairs of V-8 cylinder-heads are required, one being used on the LaSalle and Series "75" engine, the other on the Series "60" and "65."

Continental Motors has introduced a new L-head six-cylinder series known as the Model M with seven main bearings and 12 integrally forged counterweights with displacements of 271, 292, and 330 cu. in. New four-cylinder models comprise the Y series of 69 and 91 cu. in., and F 400 with 124, 140 and 162 cu. in. displacements. Hercules is in production on a Model RXLD, six-cylinder, 4¾- by 5¼-in. engine for bus and high-speed truck service, besides a model QXC, six-cylinder, 3⅜- x 4⅛-in. engine, obtainable with either cast-iron tin-coated, or aluminum pistons, and with either a three- or five-gear train at the front. There is a new four-cylinder Diesel, model DOO, 3¾ or 4 by 4½ in. The 4-in. bore engine is available with dry cylinder liners. In keeping with the trend toward higher octane fuel, the compression ratio of the Hall Scott Model 180 has been changed to 4.78:1 from 4.6:1 for city bus service, and to 4.86:1 for intercity use. With 5.6:1 compression ratio for use with ethyl fuel, 195 hp. at 2200 r.p.m. are obtained over the normal 183 hp.

On engine mountings, the Studebaker President engine has been dropped 1¾ in. and moved forward 3½ in. and the Nash 3 in. The Chrysler-built units utilize a new spool-shaped rear floating-power mounting. To suit the new frames and to provide interchangeability among the various models, the Cadillac transmission extension supports have been moved further back. Softer rubber mountings are used in the Packard-6 and -8. Compression-cup type motor mountings in the Lincoln Zephyr permit dispensing with the former stay rods.

Cylinder and Valve Gear

The Graham cylinder-head is now secured to the block by 33 studs, 13 more than last year. A reinforcing rib encircles the top of the cylinder barrels of the Packard Super-8. All Hall Scott engines now have wet liners, heat-treated, and with a hardness of 450-500 Brinell.

Valves remain pretty much as they were, all the users of austenitic steel continuing with its use. The Wilcox-Rich self-adjusting hydraulic tappet has been incorporated in the Lincoln Zephyr, Cadillac-16, Lycoming commercial, and in the White "pancake" engines. For greater strength and heat dissipation, the Chevrolet exhaust valve blends into the stem with a long taper instead of a radius, and the depression in the head is smaller. The guide no longer projects into the gas stream, and the water jacket extends down further over it. The valve springs are redesigned for greater capacity (maximum load 111-119 lb. vs. 94-102 lb.) and eliminate the former tappet springs, retainers, and their hold-down bolts and lock washers.

The camshaft timing sprockets on all Cadillac-8s with Morse non-adjustable side-guide chains of ⅜-in. pitch and 1¼-in. width, have stub teeth, whereas the crankshaft sprocket remains unaltered. The tappets on the 16 are of the mushroom type. The Packard-6 and -8 have new mushroom tappets with full pressure lubrication. The camshafts in these engines are hardened by the Tocco process. The DeSoto intake valves open and close 8 deg. earlier, 8 deg. before top dead-center and 42 deg. after bottom dead-center. Link Belt non-adjustable chain, ½-in. pitch and 1-in. widths are used on all Buicks, Grahams, and Oldsmobile-8s. The same pitch but 1¼-in. width is used by Willys. The Oldsmobile-6 has a Whitney ½- by 1-in. chain. In all instances the camshaft gear is semi-steel.

Piston and Rings

The tin-plated Autothermic aluminum-alloy piston introduced on the 1937 Packard-6 has been extended to their whole line. The top compression rings in senior engines are of electric-furnace molybdenum iron. Three compression and one oil ring are used in the 12, whereas two of each are in the Super-8 pistons. The Studebaker Lynite pistons are also tin-plated. The Buick "turbulator" piston has a special dome head, permitting higher compression and affording combustion control, along the lines of an L-head engine. Turbulence is induced at the end of the compression stroke, and the main portion of the fuel charge is brought closer to the spark-plug. The "40" engine develops 107 hp. and the "60," "80," "90" develop 141 hp., an increase of 7 and 11 hp. over 1937. The horizontal slot between the ring belt and skirt on the right

side and a T-slot at the left, together with cam grinding, permit the piston to be fitted to the cylinder barrel at normal room temperature (70 deg. fahr.) with approximately the same clearance as allowed for cast-iron pistons. The two pistons each weigh 0.911 and 1.107 lb. as against 0.859 and 1.141 lb. last year. The Cadillac-16 has low-expansion aluminum-alloy (Lynite and Bohn) anodized-finish, T-slot pistons. There are two broached grooves inside each wrist-pin bearing running the full length of the boss. One 3/16-in. oil ring and one 1/8-in. and one 3/32-in. compression rings are used.

The general trend in ring combinations is toward narrower compression rings and somewhat wider oil rings. The spring-backed compression and oil ring is receiving more recognition for installation in new engines, the latest adherent being Willys with the Sealed Power Sta-tite. The latter company has developed an iron and manganese-phosphate coating under the name of Granoseal to eliminate hot and cold scuffing. The solution used in this process consists essentially of ferrous phosphate and ferric phosphate in the ratio of 3:1 dissolved in phosphoric acid and carrying sufficient manganese dioxide to maintain the ferrous-ferric ratio. The reaction taking place is the formation of iron phosphate from the phosphorous ions in the solution with a simultaneous freeing of hydrogen from the phosphoric acid. American Hammered Piston Ring has brought out a modification of its wide-chambered oil ring in placing a 16-18-deg. bevel at the top of each land. The double-beveled surface provides a narrow cylindrical bearing area around the lower edges of the lands, providing quick seat-in, and the construction permits the ring to ride freely over the oil on the upstroke while adding to the scraping action on the downstroke.

Buick "Turbulator" Piston Providing Improved Combustion Control.

Connecting-Rod

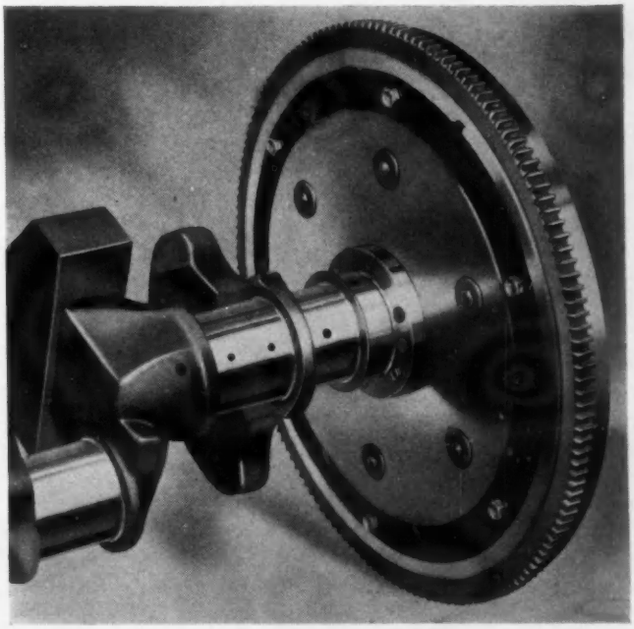
Buick connecting-rods have a new non-crystallizing babbitt bonded directly to the steel of the rod and cap which has the characteristic of becoming dark in color with use. Shims are provided to allow adjustment without filing. Hudson connecting-rods are heavier for increased rigidity. The rod and cap are now machined to a finish after they are assembled completely. Hall Scott engines are now using copper-lead bearings in the connecting-rods and cadmium silver in the main bearings.

The Cadillac-16 wristpin is locked in the connecting-rod and oscillates in the piston. A lock screw in the small end of the 6 3/8-in. long rod of manganese steel passes through a notch in the wristpin. The 2-in. crankpins permit withdrawal of the rod and piston from above without the necessity for an angle split bearing. The short stroke reduces the inertia forces of the reciprocating parts and, with the lighter power impulses, the 2-in. pin gives ample rigidity and reduces rubbing speed.

Crankshaft and Case

A 1/8 in. larger crankshaft, now 2 3/8 in., is used by Graham. Hudson obtains a greater smoothness by reducing the weight of the metal mass in the vibration damper, decreasing the periodic vibration. Cadillac V-8 engines are equipped with the "syncro-flex" flywheel, which aims to cut down the noise level by suppressing vibration due to a periodic crankshaft bending at critical speeds. The cast-iron flywheel rim is attached to the crankshaft by a steel disc acting as a web which is rigid torsionally but flexible in bending. Damping plates are spaced on each side of the web and are secured with it to the crankshaft flange, but the plates do not extend to the flywheel rim. Near the outer periphery, six bronze washers are located between the web and plate at each side and are centered by studs passing through holes in the damping plates and a large clearance hole in the web. Springs maintain the assembly in frictional contact. This construction permits the rim to run in a truer circle regardless of crankshaft deflection. When bending occurs, the damping plates absorb energy and dampen the motion. The Cadillac-16 crankshaft with nine main bearings is a high-carbon forging with eight integral counterweights. Each of the eight 2-in. diameter crankpins, which are in the usual 2-4-2 arrangement, carries two side-by-side connecting rods. The main journals are 2 1/2-in. in diameter. There is a 5/8-in. journal-pin overlap. A rubber torsional vibration damper is included in the pulley assembly at the front.

A new asbestos composition seal similar to the one introduced on last year's Oldsmobile is used on the rear main bearing of all Cadillac V-8s. Chevrolet provides a leather seal with a coiled tension spring around the hub of the harmonic balancer pressed in the front crankcase cover. The Cadillac-16 employs an engine-suction type of crankcase ventilating system. During part-throttle operation air enters the front



Cadillac Syncro-Flex Flywheel. Damping Plates Are Located Each Side of the Flexible Web and Insure Smoother Running.

and rear valve chambers through a pipe between each air cleaner and the valve chamber cover and through the filter-protected oil-filler pipe directly into the crankcase. It is then drawn up by a pipe extending from the crankcase up through the rear valve chamber and cover by a vacuum line to the intake manifolds. A 3/32-in. restriction in the pipe prevents excessive withdrawal. When the throttles are opened, a reversal takes place in the connecting pipes, air now entering the crankcase solely through the oil-filler opening.

Lubrication

Cadillac has removed the oil filter and jiggle pin from the hydraulic valve silencer supply. Better accessibility and increased road clearance have been obtained by moving the oil-pan drain to the left side on all 8s. A flexible oil-level dip-stick supersedes the rigid type. The 16 oil filler is in the center of the front valve chamber cover just behind the generator and accessible from either side. The Buick oil filler is now located at the top of the rocker-arm shaft cover. Oil filters are now standard on the Packard-6 and -8. The Purolator is used on all Packards and Chrysler units. The Float-O inlet to the oil-pump has been adopted on the Studebaker President motor.

Cooling System

There is a trend toward larger fans with a decreased rotative speed. The Packard-12 has a 22-in. four-blade fan rotating at 0.969 of crankshaft speed. The gravity flow through the radiator is 38 gal. per min. Radiator shells on both the 6 and 8 have been increased in depth to aid cooling further and larger capacity pumps are used on all Packards. All have nearly full-length water jackets, as well as full-length thermostatically operated radiator shutters. The gravity flow through the 6 and 8 radiators is 37.2 and 38.5 gal. per min. and both use 18-in. fans. Water-pump circulation is 36 gal. per min. at a car speed of 40 m.p.h.

The water pump on the Studebaker President has been moved to the front of the block. This engine and Graham's have incorporated the internal water-distributing manifold for spray-cooling the exhaust-valve seats. Chevrolet's ½ in. longer water-pump assembly permits a greater length of the copper-graphite porous bushings. The packing rings are fed from the oil reservoir by a rod of similar composition but of different porosity. The fan blades are separate stampings welded and riveted to a stamped spider.

The New Departure unit fan-and-water-pump bearing, sealed-for-life, is used by Cadillac, Ford, Graham, Oldsmobile, Packard, Pontiac, Willys, and General Motors Truck. Hudson is using a synthetic rubber-and-cork combination to seal the pump, and on the 8, additional cooling capacity has been secured by a more efficient core design and by increasing the frontal area. Pontiac has increased its pump capacity. The Schwitzer-Cummins packless pump with synthetic-rubber carbon seal is used by Ford, Studebaker, White, and Hercules.

The Lincoln Zephyr, Cadillac, and LaSalle radiator cores, now lower and wider, are of greater capacity and more efficiently covered by the fan. In increasing the LaSalle grille 2 in. in width, the area of the grille openings has been stepped up from 194 to 215 sq. in.; on the Cadillac "65" and "75" from 257 to 269 sq. in. The 16 has two independent circulating systems and the two water pumps are driven by a quadrangular double V-belt drive. The bottom radiator tank has two outlets each connected to a pump. The impellers are held on their shafts by a press fit. The shutter thermostat is located in the rear of the top tank instead of in the front, increasing accessibility and protecting the thermostat from cold air blasts which might cause irregular operation. A seven-blade Hayes fan with asymmetrical spacing rotates at 0.95 of engine speed. It is entirely unblocked over the upper half due to the low block contour.

United Parcel Service is building a 350 cu. ft. delivery unit for its own use with steam cooling, the condenser unit being placed above and close to the roof.

Fuel System

Graham is using a Marvel triple-venturi carburetor in which a 20 per cent larger opening can be utilized and greater suction placed on the jets. The LaSalle and Cadillac-16 are equipped with Carter carburetors and, to improve hot-starting, the metering jet is air-cooled. Nozzle changes to give a more accurate calibration of fuel-air ratio improve the engine "feel" during part-throttle operation. The action, when cold, is aided by a slower decrease in choking as the engine warms. The "60," "65," and "75" are Stromberg-equipped, and a vacuum-actuated economizer is employed instead of the former one operated by the throttle, thus enriching the mixture whenever power is required even though the throttle is not fully open. With the new construction, the fuel flow increases at small manifold depressions, enriching the mixture whenever the engine is pulling regardless of throttle position. This arrangement makes possible the use of a leaner mixture for a light-load operation with resultant economy. The Carter-equipped deluxe Terraplane uses a single-throat manual-choke carburetor also with a vacuum-controlled metering pin. Chevrolet provides an improved vent in the air horn for balancing the pressure between it and the bowl.

The Cadillac-16 engine provides two independent fuel pumps supplying the two carburetors from a 26-gal. gasoline tank. Fuel is drawn forward through a single pipe on the outside of the right frame member. Near the front of the engine a branch passes to the right pump, the main line continuing forward over the shock absorbers across the front side of the front cross-member and back on the left side of the radiator to the left fuel pump. Each pump is actuated through a push-rod by an eccentric on the adjacent distributor drive-shaft. There is an interconnecting line between the pump-to-carburetor lines so that either pump may supply both carburetors. The 1⅛-in. Carter carburetors are of the dual down-draft type with climatic control. The fuel mixture passes downward to an exhaust-heated division chamber fed by the two center exhaust ports of each block. The four legs of each intake manifold are so designed that no two successively firing cylinders of a bank draw from the same manifold leg. The Buick and Oldsmobile carburetors provide a richer mixture during gear-changing intervals with the automatic transmission when the engine speed is likely to drop with an open throttle.

The Buick throttle rod is tubular to prevent vibration and is secured to the accelerator lever by means of an oblong piece of rubber-impregnated fabric which eliminates any linkage backlash as well as to insulate the pedal and car interior from noise and heat. The throttle return spring is anchored to an angle piece on the insulated hinge which is provided with three adjusting holes. A ⅜-in. heel ledge is located at the bottom edge of the accelerator pedal. Automatic chokes of all Studebaker engines are now mounted directly on the Stromberg carburetors, eliminating the previous linkage. The choke thermostat is arranged so that hot air is drawn from a pipe extending through the manifold, making the thermostat more sensitive to engine temperature. The President engine is equipped with the aerotype carburetor. An 18 per cent increased capacity is provided in the Plymouth fuel pump.

The Schwitzer-Cummins supercharger used on some of the General Motors coaches is a complete self-contained unit with the engine water pump, its own oil pump and oil reservoir, and is driven by three V-belts. For operation at a 13,000-ft.

elevation, Sterling has equipped some of its Diesel trucks with a Schwitzer-Cummins supercharger in which there is a fixed bypass between the inlet and outlet lines which can be locked to the desired position to regulate the air volume for high altitude or sea level. The supercharger runs at 1.75 of engine speed and weighs 60 lb. It delivers 512 cu. ft. of air per min. against a pressure of 5.8 lb. per sq. in. to equal sea-level pressure at the elevation noted.

The Buick tank-filler spout has been moved to the left side. The filler is concealed beneath a small spring-controlled door fitted flush in the top of the rear fender. Any overflow when filling runs to the ground.

Exhaust System

The Hudson-6 and Terraplane have straight-through type mufflers with toning chambers. The previous heat baffle plate has been eliminated in view of the cooler operation. Buick likewise uses straight-through resonance-type mufflers, and the 2-in. diameter tail pipe eliminates any possible back pressure.

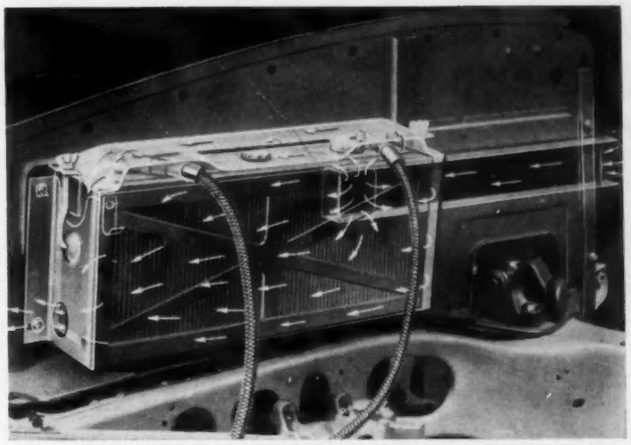
On the Cadillac "65" and "75" the exhaust take-off is now on the left side to suit the relocated exhaust system which runs outside of the frame rail. At a point close to the rear anchorage of the frame X-member, the pipe runs inside of the frame and has a flexible connection to the transverse muffler back of the gasoline tank with a short tail pipe located at the right. Due to the lowered frames, the mufflers are mounted transversely at the rear. On the "90" the two exhaust manifolds discharge into a cross-pipe which connects with the exhaust pipe leading backward outside of the left rail. On the "60" it is on the right side of the chassis and runs within the frame rail connecting to the rear transverse muffler with the outlet at the left. A separate expansion chamber is placed in the exhaust pipe ahead of the muffler on all models, dispensing with the former exhaust spreaders.

Graham is now using a dual arrangement of exhaust manifolds with three cylinders exhausting into each member to obviate overlapping. A long single-unit muffler replaces the tandem type on the Packard-6 and -8.

Electrical Equipment

Hudson and Terraplane are using Champion J-8-A plugs set at a 0.032-in. gap. The higher capacity coil permits satisfactory operation up to 0.040 in. Additional guards over the torque arm bracket and the right-hand-drive steering gear hole in the frame protect the distributor from road splash. Improved combustion control with the Buick "turbulator" piston permits a 6-deg. flywheel initial timing advance over 10 deg. in 1937 on the larger engine. The Cadillac-16 engine employs two distributors one of which makes and breaks the primary current for both iron-clad coils. The left-hand distributor has two breaker arms, the fixed one making and breaking the primary current for the left-hand coil and cylinder bank. The other arm is adjustable and acts as a breaker for the right coil and cylinder bank. Its advance mechanism thus determines the spark advance for both blocks. The left distributor rotor furnishes high-tension current to the left block only. The right-hand distributor contains no breaker mechanism and its advance mechanism merely keeps the rotor in correct position. The two coils are mounted above the distributors on the radiator thermostat housing.

Generators have been stepped up in output due to improved ventilation in most cases with usually a 25-amp. minimum output. Chevrolet, Graham "97," Oldsmobile, Pontiac, and Packard-6 and -8 use high-output third-brush type generators, and vibrating voltage regulators. Voltage regulation is used on Lincoln Zephyr and all Hudsons, being optional on the Terraplane. The Cadillac-8s and Buick continue with the high-output split-field generator with voltage regulation, but the generators are now suspended on a swinging bracket on the side of the cylinder-block. The Cadillacs "75" and "90" and Studebaker President have straight shunt generators with external current and voltage regulation. To improve the low-speed charging rate, generator speeds have been stepped up on the Cadillac "60" to 5450 revolutions per mile over 5230 in 1937, and to 5725 revolutions per mile on the "90" over 4270 in 1937. The generator drive-belt life is increased by larger pulleys on all Cadillac V-8s. The 16 generator, rotated at 1.95 of engine speed, is driven by the dual-belt fan pulley assembly which also incorporates an inner conical drum containing at the rear a rubber driving ring against which the generator driving wheel is pressed. The fan-pulley combination is mounted by a bracket attached to the front of the distributor and fuel-pump support by two studs, one of which is in a slotted hole in the bracket permitting adjustment of belt tension. The generator front cover is secured similarly to the fan mounting bracket to adjust frictional contact between the rubber ring and the driving wheel. New Departure sealed bearings are used in all Cadillac generators.



End-To-End Delco Battery. The Oldsmobile Installation Provides a Duct to Convey Cool Air to the Battery Compartment Alongside the Engine.

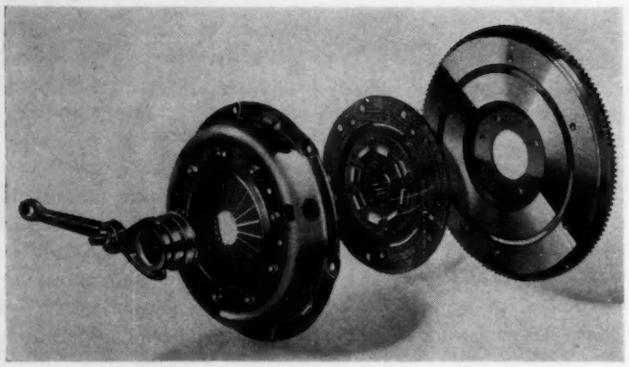
Chevrolet provides a mechanical shift drive employing an overrunning clutch. The "starterator" is no longer used, but the throttle is now partially opened by a lug on the starter pedal shaft engaging a corresponding lug on the accelerator rod. Four-pole starting motors are used on the Cadillacs, except the 16 which is of the six-pole type and drives the starter pinion directly instead of through reduction gearing. The Commander and Studebaker-6 are using a pushbutton on the instrument panel. Buick's vacuum starter switch for coincidental starting is now incorporated in the Delco-Remy automatic-choke unit. A latch in connection with the vacuum mechanism allows the contacts to open only after the engine vacuum has built up to 5 in. of mercury and holds them open until it drops to ½ in. and the throttle is closed. This range guards against untimely starting motor engagements.

With the increased use of under-hood battery mounting, a long thin end-to-end unit has been developed by Delco for Pontiac, Buick, and Oldsmobile who mount it above the frame rail. New vent caps have been designed to prevent acid spray and splash. The short connecting links are recessed in the cell covers and are sealed over with wax to guard against accidental short-circuiting. The starting cables are shortened considerably. Oldsmobile provides a duct which directs cool air to the battery compartment. Lincoln Zephyr has the under-hood mounting. Except in the 16, Cadillac mounts the battery under the right side of the hood alongside the frame rail. A battery filler is offered as an accessory.

A bottle holding distilled water is mounted on the dash, one for each cell, to which it is connected by a supply and vent tube. A check of battery condition is possible since a rapid drop of water level in one bottle would indicate trouble in the corresponding cell. Sufficient acid from the battery will rise to the reserve bottle through the vent tube to lower the freezing temperature of the distilled water.

Clutch

Chevrolet's new clutch employs a diaphragm spring which has the shape of a dished washer with 18 integral inwardly pointing tapering fingers. When the clutch is engaged the fingers are flat and the entire rim of the spring bears against



Chevrolet Clutch. The Diaphragm Spring Replaces Numerous Parts Previously Used, Decreasing Pedal Pressure During the Stroke.

the pressure plate giving uniform load distribution. Pressure of the throw-out bearing on the fingers causes a cupping action for release. Rings are provided at each side of the diaphragm to act as fulcrums. Pedal pressure decreases during the stroke.

The Hudson-6 and Super-Terraplane use a larger clutch with 12 pressure springs in place of the previous 9. The driving-disc circumferential damping springs are heavier on all models, and the friction discs are cupped to provide additional friction. On the 8, 15 springs are used instead of 12, the three additional ones being nested within the conventional springs.

The Cadillacs use a single-plate semi-centrifugal clutch. On the 8s, 8 vibration-dampening springs are used instead of the former 10 to decrease the spinning weight. The spring wire size has been increased as well as the size of the slots in the clutch disc, to carry the engine torque with fewer springs. Due to the smooth flow of power in the 16, no spring dampers are used. All Chrysler products are equipped with a baffle pressed in the clutch pressure plate to reduce the possibility of oil or water vapor reaching the disc surfaces. A grooved facing aids cooling and breaks any vacuum. Plymouth is using two woven facings in place of one molded and one woven for improved smoothness of engagement, which smoothness also is aided by an integral-type cushion. The Studebaker President employs the Long semi-centrifugal clutch.

The centrifugal bursting strength of linings has shown considerable improvement, fostered by increased speeds and the tendency to do more shifting at high speed. Long has established a minimum bursting strength of 7000 r.p.m. hot. Its laboratory procedure is to attach the linings to a disc, "soak" in an oven to 450 deg. fahr. for 15 min., remove quickly and put on a spinning fixture with the speed gradually increased until the linings burst. Smooth engagement has necessitated the establishing of cushion deflection curves with high and low limits which, when established experimentally, are used in checking production units periodically. Chatter has called for harder rubber mountings in some cases, and the design of the pedal linkage is critical. One manufacturer is considering the remounting of the clutch pedal on the powerplant.

The clutch-pedal assist spring introduced on last year's Oldsmobile is now used by Pontiac and Buick. This part consists of an over-center return spring. The Buick clutch-release bearing retainer is made of steel stampings and hardened where it contacts the release yoke, the latter being hardened on the face of the fingers. The sealed-for-life release bearing has an oil-soaked wool felt between the small bushings in which the steel balls ride.

Transmission

The automatic self-shifting transmission recently offered as optional equipment on the Oldsmobile-8 is now available on the -6, as well as the Buick "40." It consists of a single-reduction planetary unit in series with a double-reduction set so that either of four ratios is available. A sliding direct and reverse gear set is placed ahead of the planetary system. Each of the two planetary brake bands is tightened by a nest of coil springs which can be compressed to free the drums by a hydraulically operated piston under the influence of a centrifugal governor, a throttle control, and a regulating valve. The selector lever under the steering wheel has four positions – "reverse," "neutral," first and second gears in the "low range," and a "high range" for third and fourth. The clutch is used only in starting and, with the selector lever in "low range," the car moves forward in first gear; at approximately 6 m.p.h. the transmission shifts automatically into second. At any speed above 6 m.p.h., upon moving the selector lever to "high range," the shift to third gear is then made without disengaging the clutch and, at 23 m.p.h. or higher, fourth gear (direct) is engaged. For fast acceleration or added hill-climbing ability at any speed between 23 and 55 m.p.h. complete depression of the accelerator pedal brings third speed into action until a speed of about 55 m.p.h. is reached when direct engages. At any time that the throttle is closed partially, third gear engaged, and the car speed over 23 m.p.h., the transmission automatically shifts into fourth. When in the "high range," the transmission automatically selects first, third, or fourth speeds according to the speed of the car or the position of the throttle. Transmission ratios are 3.16:1, 2.23:1, 1.42:1, and 1:1 with 3.20:1 for reverse. The axle ratio is 3.55:1 as against 4.375:1 for the conventional transmission. Buick uses 3.615:1 as against the 4.40:1 standard, and shifting speeds vary slightly.

Gear noises are suppressed with remote control since the conventional lever is absent to telegraph them into the passenger space. Hudson, recent pioneer of remote control, continues with a selective automatic shift wherein, if upon shifting with the "electric hand" there should be an abutment of the gears in going into low or reverse, the selector lever automatically returns to neutral. The automatic-clutch circuit-breaker switch has been improved for smoother shifting and to compensate automatically for any possible clutch-facing wear. A balanced-vacuum cylinder located at cylinder-head level for clutch actuation provides closer control, and the former pendulum device has been eliminated.

Pontiac's optional gear-shift control comprises a lever under the steering wheel projecting to the right and mounted on a control tube slightly ahead of and parallel with the column, to the lower end of which is welded a lever with a ball joint. Under the toe-board a rod runs to a lever depending from a single oscillating transverse shaft in the transmission cover-plate, on the inner end of which is mounted a cyanided spherically ended lever. Within the control tube slides a selector

control rod, raised or lowered as the control lever is rocked for selection. At the base of the control rod, a flexible cable conveys the motion to a selector plate which transfers the shifter motion from the cyanided lever to either shifting bar. Cable adjustment is made beneath the inspection plate in the cover where a threaded stem from the cable passes through a lug in the selector plate with a nut on each side of it.

Cadillac and LaSalle's "syncromatic" transmission control consists of a short lever projecting to the right from a housing just below the steering-wheel hub. It is pivoted within the housing on the left side of the column, flattened at this point, and provided with a slot to clear the column. By rocking the ball end parallel to the column axis, one of two concentric shafts is engaged, both extending down the left side of the column and nearly parallel to it. Below the toe-board each shaft carries a lever from which a shifter rod runs to a lever on the transmission. One lever is secured to the oscillating shaft controlling the high and intermediate positive clutch whereas the other is on the shaft controlling the low and reverse sliding gear. On right-hand drive cars the lever extends to the left. The Cadillac countershaft has been increased in diameter, and the number of needle bearings supporting the gear cluster increased from 26 to 28. The second-speed countershaft gear width has been increased ⅛ in.; the main shaft spigot has been reduced from 1⅛ to ¾ in. diameter, and the former caged roller bearing is superseded by one without a cage. The full complement of rollers maintains the capacity but permits of more material to support the integral constant-mesh gear. The synchronizers have been narrowed 1/16 in. to increase the unit pressure and, thereby, the gripping power of the clutches. The synchronizer drums have been lightened and strengthened by the use of a tough corrosion-resisting aluminum alloy. With the oscillating control shafts, the interlocking balls engage notches in the inner shifting-lever hubs.

The Evans vacuum gear shift is used optionally by Graham, Nash, and Studebaker. A small shift lever about 5 in. long is located slightly to the right of the steering wheel and just below the instrument panel. It can be operated manually should anything happen to the vacuum line or should the engine be dead. Graham uses an Evans diaphragm mounted below the left-hand side of the transmission, to operate the cross-over between the two shift rails. Studebaker performs this function manually with a positive linkage. Nash locates the diaphragm at the right side of its transmission, running a vacuum line up to the dash selector unit. A vacuum cylinder normally shifts the gears.

The Warner T88 transmission used by Studebaker and Graham has the countershaft to the right of and in a horizontal plane with the main shaft, the shift rails and forks being to the left. Should this transmission be desired for direct manual operation, the short control pedestal permits a low floor and without sacrifice of road clearance. The control cover on the left permits a number of possibilities for mechanical operation as well as Bendix or Evans control.

Studebaker, Lincoln Zephyr, and Chrysler-built units provide a new blocker ring to the high-speed side of the synchronizer unit to facilitate shifting into high gear. Warner Gear has developed an overdrive unit in which the operator can get out of overdrive whenever he desires, a new member being sandwiched between the main transmission and the overdrive unit affording an inertia release to the sun gear which is normally held fixed.

Hudson provides a heavier and ribbed transmission case, whereas involute splines are used on the shafts. A circular lock-ring has been added to the second-speed main-shaft cup gear, and additional sediment space is provided at the bottom of the case. As special equipment, Chevrolet offers a brass plug with a permanent-magnet insert to substitute for the regular oil drain plugs of the transmission, rear axle, or engine. Improved shifting on the Buick "40" has been obtained by the use of two-lobe type synchronizers retained and cushioned by hexagonal springs instead of separate detent cushion springs. The countershaft gear bearings are held in place by a perforated bearing spacer, improving the flow of lubricant to the bearings and also tending to clear and hold particles of dirt by centrifugal action. The bearing area of both shifter forks has been lessened to reduce the possibility of noise. Buick uses self-locking shifter yoke setscrews. The rear bearing retainer is provided with heavy section walls, amply ribbed to maintain alignment of the universal-joint and ball assembly and to absorb rear-axle thrust. The clutch shaft bearing now protrudes about ⅛ in. from the case to pilot the flywheel housing on it. The Chevrolet ¾- and 1-ton trucks have carburized gears. First and second gear ratios are faster on the Willys.

Propeller-Shafts

The sealed-for-life rubber-mounted propeller-shaft bearing introduced by Oldsmobile last year is also used by White and General Motors truck. All passenger cars are now equipped with needle-bearing universal joints, the last to come into line being the Willys with the Universal Products 4100 Series joint where slip is taken by a ball-and-trunnion construction. On the long-wheelbase Hudsons a 3½-in. propeller shaft is used. The Chevrolet 127-in. wheelbase taxicab utilizes a short jackshaft ahead of the half-ton truck torque tube and propeller-shaft.

Rear Axle

Hypoid rear axles have been adopted by Graham, Buick "80" and "90," Cadillac "65," "75," and "90," and Lincoln Zephyr. The new Cadillac hypoid axles follow the same design as the 1937 LaSalle and Cadillac "60" except that threaded sleeves are used in place of the shim adjustment for the differential bearings; the pinion shaft bearings are Timken which are preloaded at assembly and adjusted with shims. A third change is in an increase of spiral angle from 45 to 50 deg. The housing "cover" is spherical in shape, arc-welded to the housing in a complete circle as viewed from the rear and spotwelded at the front to the inside of the housing besides being held there by the carrier bolts. To reduce overhang, the wheel-bearing on the LaSalle and Cadillac "60" axle has been moved out ⅜ in. and, in the larger axle, ¾ in. To permit wider bodies, the Series "60" and "65" rear treads have been increased to 61 and 62⅜ in., respectively. Buick uses the Hyatt barrel roller bearing at each side of the differential. The spline size and neck diameters of the axle shaft have been increased, and the torque tube strengthened at the entrance to the carrier. The Spicer axle used by Graham has Timken bearings in the rear wheel in place of ball bearings and the tread has been increased to 61¼ in. Packard-6 and -8 have changed from ball bearings to single-row Timken bearings in the wheels. The Marsden thread is used on all Packard pinion shafts. The Lincoln Zephyr hypoid gears have an overhung pinion, Timken mounted. Hupmobile is equipped with Spicer axles as heretofore. Hudson has increased its tread by 1½ in. on all models to permit oversize tires without interference while Chevrolet has increased it 1 7/16 to 59 in. for increased rear seat width. Taper splined parts made by the Barber-Colman hobbing process are used by Wisconsin Axle and Timken-Detroit. The Chevrolet axle has a double-eared bracket welded to the front side of the housing at each side, and an eye bolt passes through same and a lug on the top of the redesigned spring seat. A rubber bushing and washers at each end insulate the lug from the

bolt, and a steel sleeve around the bolt acts as a spacer between the bracket ears.

Brakes

Johns-Manville has developed a molded lining quite immune from fading that is used on the Chrysler-built units, a close relation of the same formula being used by Ford. It has been found possible to make 15 to 20 stops in rapid succession at high decelerative rates of 18 to 20 ft. per sec. per sec. from 70 m.p.h. without measurable fading in the friction of the lining. Continuous service of 30,000 to 50,000 miles is claimed.

Dodge and DeSoto brakes have been stepped up from 10 to 11 in. in diameter, the shoes are heavier and stiffer, and the drum more rigid. This improvement and better cooling are found in all centrifuse drums in view of a high thin rib near the open end in place of the former low wide rib. The drums are closer balanced. Hudson continues alone with the stamped drum. The Cadillac "90" brakes are similar to the "75" except for the use of machine-ribbed drums for better cooling with the heavier car. Pontiac incorporates wheel ventilating slots to provide 80 per cent more air for quicker drum cooling. With the elimination of the swiveling spring seats and caps, Chevrolet runs the hydraulic brake line along the rear of the axle housing. Linderman Devices has developed a residual volume valve for the hydraulic master cylinder to provide quicker release. The entire brake line, including the master-cylinder area, is left wide open until the last ¾ in. of pedal travel. A control valve provides an adjustment for the ratio of front and rear braking. Nash now offers the NoRol as optional equipment, continuing similarly on the Hudson and as standard on Studebaker. All Studebaker linings have been widened to give a 16 per cent increase in surface, the President having 169 sq. in. and the Commander and Studebaker-6, 150 sq. in.

All the Chrysler-built units have a pistol-grip hand-brake control under the dash and to the right of the driver where it also can be reached by the front-seat driver in case of emergency. Buick and Studebaker provide new cable hook-ups for the parking brakes. The Packard cable is insulated completely in rubber. The Chevrolet hand-brake sector moves in the opposite direction to the lever, being pivotally mounted on the brake rod clevis pin and tied by a link to the frame bracket. The pawl rod is within the tubular lever and the curved release finger turns down toward the back. Midland has developed a house trailer vacuum-brake system with the hand control valve on the steering column below the wheel to actuate the self-energizing steeldraulic wheel brakes.

Front Axles

The upper steering-knuckle bushing in the Plymouth has been replaced by needle bearings. The Cord double-row ball bearing mounting in the front wheel has been redesigned to use a pair of Timken bearings. Hudson has adopted oil seals at the spindles, of a synthetic rubber and cork compound.

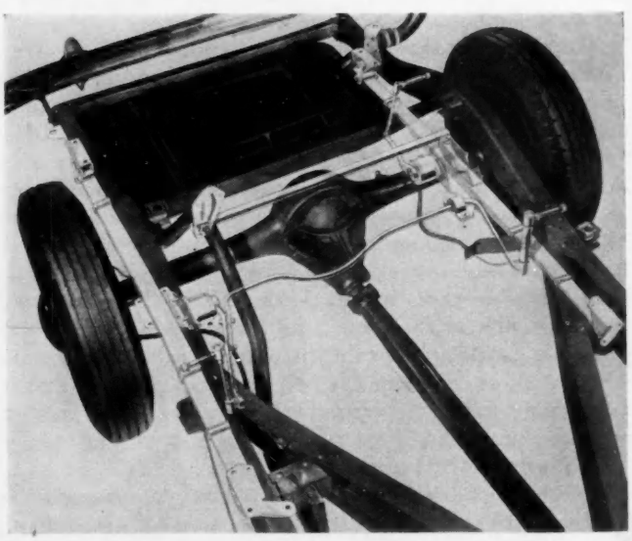
Wheels, Rims and Tires

The Kelsey-Hayes Nash disc wheels are provided with three perforations near the rim for the installation of emergency skid chains. The Motor Wheel Packard-6 and -8 disc wheels are similar to the unchanged Oldsmobile wheels except that they have four short legs at the rim in place of six. The Buick disc contour is slightly changed to accommodate the new-design hub cap. Chrysler continues with plain discs except for three depressions at the rim for chains. The Hudson-8 disc is distinguishable over the -6 in that it has circumferential corrugations. Pontiac's disc now has ten slots near the outer periphery in place of the previous solid web. Lateral and radial rim runout is held within 0.031 in. of total indicator reading. This runout is held to 1/16 in. on the Packard-6 and -8, Buick, Hudson, and Chrysler-built units. Static balance is held to within 12 oz-in. on Pontiac wheels and 15 oz-in. on Buick and Packard-6 and -8. Rims remain 16 in. with the exception of 15 in. on the Buick "60" and Hudson's optional equipment. The General Motors T-15-H truck dual wheels with 16 by 4.50 "E" rims have an entirely new type of center mounting with 5⅜-in. hub, 6 stud, 7¼-in. bolt circle. Hudson has located its rims ¼ in. closer to the bolt-mounting flange to reduce overhang.

Tires show no radical changes, there being a tendency toward the ribbed tread for noise elimination. Goodyear has developed a YKL tire for trucks and buses in which the rayon 275-denier single yarn (corresponding in weight to about a 19's cotton) is made into a four-ply strand, three of which are cabled. This 275/4/3 cord is about 12 per cent lighter than the standard 23/5/3 cotton. The rayon cord is treated by a special solution to bond it with the rubber. Higher tensile strength at elevated temperatures is obtained, and tests show considerably increased mileage.

Suspension

Buick's rear coil-spring suspension aims at reduced unsprung weight, elimination of interleaf friction, elimination of shackle members, and better tracking of the rear wheel with the front. The springs are attached to the frame and rear axle by a cup and bolt which anchors the end coils. The upper bolts have right-hand threads and the lower, left-hand threads. Rubberized-fabric spacers insulate the springs at the frame end. They rest on a seat projecting back from the underside of the axle housing. The axle is held in place fore and aft by the torque tube and transversely by a stabilizer link attached to a bracket welded to the axle housing on the right end and to a bolted frame bracket at the left. The link is of substantially hollow rectangular section, formed by two stampings welded together and to tubular liners at each end which are rubber-bushed. The spring rate (pounds per inch of deflection) is 150 for the "40" and "60" five-passenger sedans, 125 on the coupes, 170 on all "80" models, and 180 on the "90." The inside diameter in all instances is 5 in. A Delco 1⅜-in. direct-acting shock-absorber is attached to the forwardly extending spring seat and tilts forward to a bracket on the top of the frame rail. The main difference in this unit over the other Delco 1-in. models is in the construction



Packard-6 and -8 Rear Suspension System. Shock Absorbers, Sway Eliminator, Stabilizer Link and Cupped Spring Leaves all Contribute Their Share.

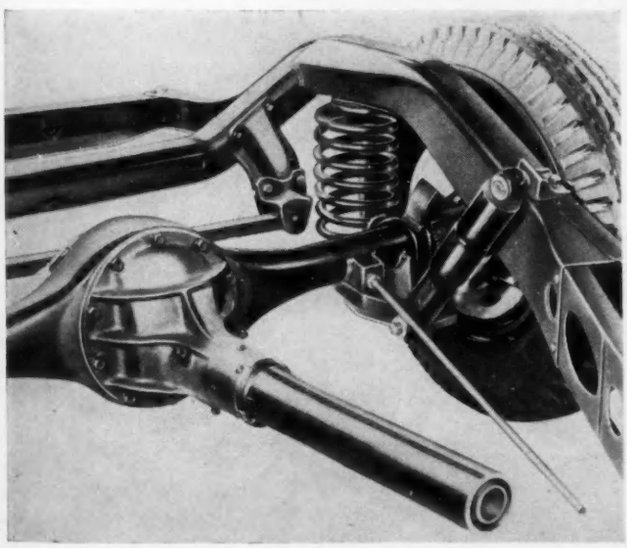
of the compression valve and the addition of an orifice plug in the rod guide.

The Packard-6 and -8 rear-spring leaves are provided with cupped depressions, some containing buttons of rubber and the others of oil-impregnated bronze. The spring eyes have rubber bushings with an integral flange at one end. A flange is formed at the other end when the shackle is drawn up. A tubular stabilizer link runs from the left rear-spring pad to the end of a vertical bracket at the right of the frame. A Houdaille shock-absorber on the 8 and Delco on the 6, both of the two-way lever type, are axle-mounted and so that the arm of the left unit projects to the rear whereas the right projects to the front. The latter unit connects with the offset end of the sway eliminator bar which at its left side is bracketed in an axle-mounted rubber bushing. This construction permits the use of more flexible springs as they are somewhat relieved of starting and braking torque.

The spring liners used in the 1937 LaSalle and Cadillac "60" are now used on the larger Cadillacs. The previous spring covers, which were found to act as dust collectors, are no longer used. The center-to-center distance between the rear springs has been increased to reduce roll. Two-piece rubber bushings are used in the front spring eye and in the upper shackle bearing of the Cadillac "65," "75," and "90," the lower shackle bearing being threaded. In addition to Buick and Packard-6 and -8 using the transverse stabilizer link which last year was to be found only on the LaSalle, Cadillac "60," and Packard-12, it is now incorporated in the Packard Super-8, all Cadillacs, Chrysler Imperial, and Custom Imperial. Lincoln Zephyr has dropped the stabilizer and vertical shackles and resorted again to angle-mounted shackles. The Hudson front-sway eliminator now has increased leverage. It is available at extra cost on Packard-6 and -8. Buick has increased the diameter of the bar. Nash has adopted the direct-acting type of shock-absorber which inclines inwardly and upwardly at the front axle, Delco and Gabriel being used. Graham uses Delco direct-acting shock absorbers with the front vertically disposed cylinders just within the frame rail at each side. The Houdaille double-acting shock-absorber is to be found on Ford, Lincoln, Pierce-Arrow, Packard-8, Studebacker, Mack buses, and Diamond T trucks.

Individual Wheel Suspension

The Studebaker planar front suspension now incorporates a radius member in front of the spring instead of below, and a lever-type of shock-absorber replaces the direct-acting type. The previous "pivotal" center-spring mounting is now rigid. The tension side of the main and second leaves are ground. The Packard-6 and -8 front coil spring is now cushioned by a molded-rubber pad between it and the front cross-member. Spring frequency has been reduced from 74 to 66 oscillations per min. DeSoto and Chrysler use hardened steel bushings on the control arms which have been redesigned in the latter two cars for increased rigidity. Hardened threaded bushings have been added to the inner end of the Pontiac, Oldsmobile, and LaSalle lower control arm. Synthetic-rubber oil seals protect the bushings of all of the foregoing makes. The LaSalle and DeSoto rubber rebound cushions have been made softer.



Buick Rear Coil Spring Suspension. Torque Tube and Transverse Stabilizer Link Maintain Axle in Position.

Frame

In place of last year's I-beam X-member, Buick now has a channel section which faces outwardly to form a box section at the junction with the side-rails. The forward arms join the side-rails at a point near the dash. There is an additional channel extension to complete the box section forward which extends to the front cross-member in the "80" and "90" but, on the "40" and "60," it bends inward before reaching the cross-member to form a diagonal brace. The rearward X-member arms join the side-rails immediately in front of the kick-up, and an additional channel extension from this point forms a box section over the kick-up which is flared out on the end to form an irregular box section with the side-rail and the new tubular cross-member. The side-rails taper in depth more gradually than in the previous models from the dash back to the kick-up. A new Z-section is used as a rear cross-member on the "40" and "60." Welding is used throughout. Additional strength for convertible body frames is provided by making the frame rails, X-member, and front sub-rails of thicker stock. Oldsmobile furthermore reinforces the center of the X-member with a large plate, top and bottom.

The Studebaker X-member consists of two straight continuous channel members, considerably enlarged at the center around the propeller-shaft opening, telescoping each other at the center. The major center portion of each channel web adjacent to the center opening is offset to a central plane. With the addition of some stamped members riveted and welded to the X-juncture, an I-beam center results with "flanges" in the form of a closed hollow V, increasing the moment of inertia. The President frame is 18 per cent lighter and shows a 70 per cent increase in torsional rigidity. The Commander and "6" frame is 8 lb. lighter and shows a 300 per cent increase. Key-welding is used throughout between the side-rail and cross-members, battery hangers, in the frame rail box sections, and so on.

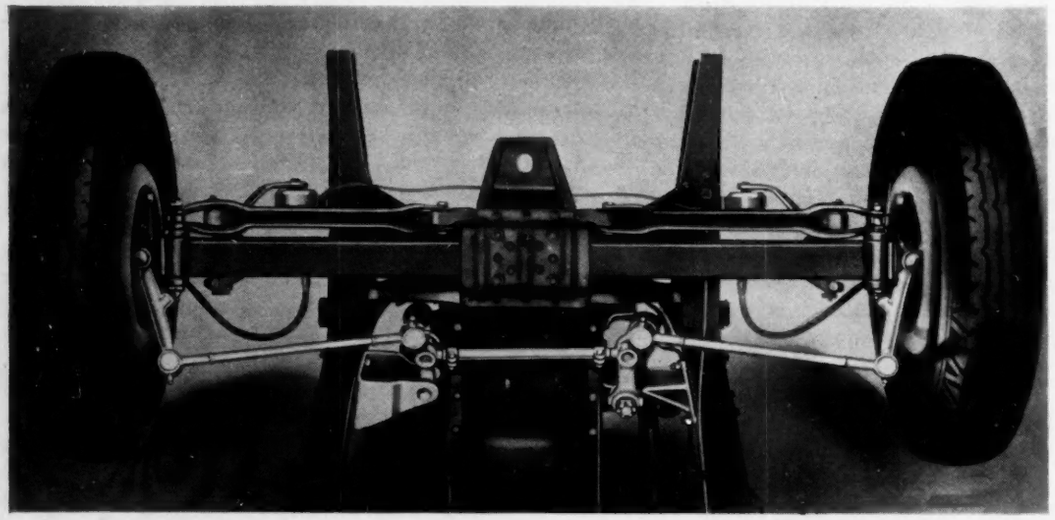
The DeSoto frame is approximately 5 in. wider at the dash and 7 in. wider where the steering gear is mounted. Side-rails have been increased from 5½ to 6-in. depth and two sub-side-rail members have been added at the dash line. The center X-member has been increased to a depth of 9½ in. Graham has added a cross-brace to the center of the X-member for rigidity. These cars have a large tubular front cross-member to which diagonal tubular braces are anchored centrally, running down from the radiator core frame. The body brackets have been located where either the X-member or cross-members join the frame rail. A light steel-truss system joins the front of the body and frame to brace the hood against weaving. The Cadillac "60" frame has been lowered approximately 3 in. by the use of a double drop frame having an 8-in. ground clearance at the center, a deep kick-up over the rear axle, and a raised front. The "65" and Fleetwood frames incorporate an X-member central junction which has been

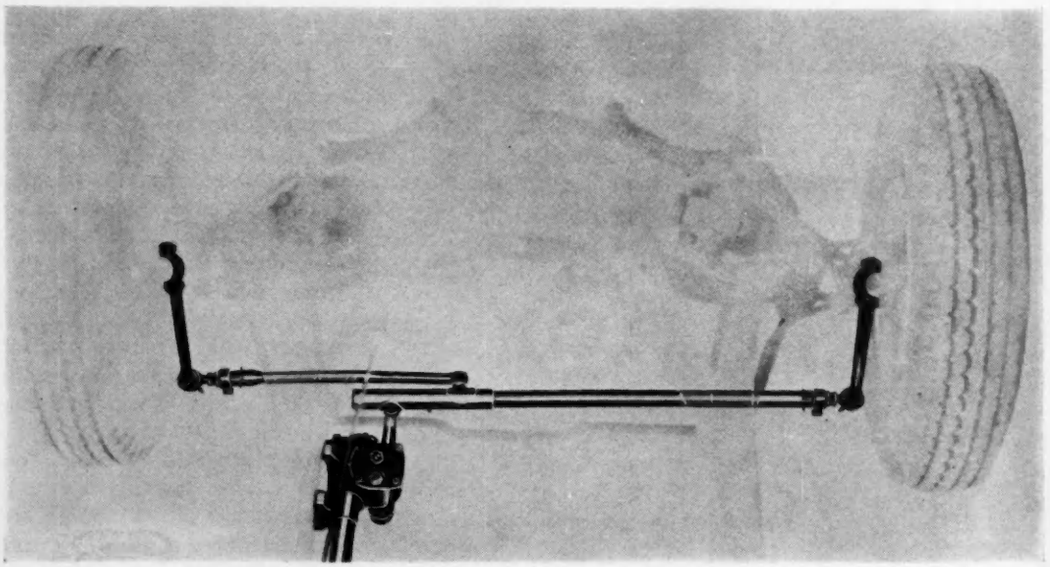
increased to 9 13/16 in. The LaSalle frame has been changed in the tipping down of its front end to a horizontal position to center the caster adjustment of the front wheels.

Control

To get exactly equal steering action on both wheels, the previous long right Studebaker drag-link has now been shortened to the same length as the left, being controlled by an idler lever actuated by and simulating the geometry of the steering-gear pitman arm. Buick has dispensed with the center frame-mounted idler lever on the "40" and "60" by utilizing a drag link running directly from the pitman arm to the right knuckle arm. The link running to the left arm extends to a ball-and-socket joint in the right drag link a short distance to the right from the pitman arm ball stud. The latter is cushioned by a coil spring at each side, whereas the left link take-off is uncushioned. The DeSoto, Chrysler Royal, and Imperial steering gears are rubber-mounted in the frame. On the Chrysler Custom Imperial an intermediate steering arm is now used with individual drag-links to each wheel. In dropping the steering gear ratio from 16.4:1 to 14.6:1 and with the use of a pitman arm of 7 in., the Plymouth overall steering ratio becomes 17.6:1 which is 5 per cent faster than last year. The Hudson steering ratio is now 18.2:1, and the drag link has been strengthened by heavier section and change in material.

The Ross twin-lever gear continues on Graham and Studebaker, the latter increasing the ratio to 21:1 on the straight-away and 26:1 at both extremes. Ross has developed the T 26 twin-lever gear, used on some of the International Harvester, Diamond T, and White models. Gemmer has adopted an external adjustment for the meshing of the worm and roller tooth for normal wear by slightly offsetting the roller from the steering-column axis. The adjustment is very close due to the resultant tangential approach. The Gemmer design, built either by Gemmer or the car manufacturers themselves, will be found on all the General Motors cars, Chrysler-built units, Ford, Lincoln, Nash, Hupmobile, Hudson, Cord, and Packard. For greater ease in steering, the Ford truck is now equipped with the roller type of gear. The Buick and Graham steering-wheel spokes are recessed for the horn ring which is also found on Chrysler. Chevrolet provides a ring, as accessory equipment, which fastens to the column below the wheel. A light beige wheel is used on the Cadillac "50," "60," and "65," and a dark-brown wheel on the Fleetwoods. The Chevrolet wheel, horn button, and column are enameled plover beige, corresponding with other interior appointments; instead of being keyed to the shaft, the wheel hub has 36 serrations so that it can be positioned as desired. The angle between the pitman shaft and the column on the De Luxe has been changed from 94 deg. 13 min. to 90 deg., making the housing alike on all models. The Master has a roller tooth on roller bearings, replacing the previous sector. For better view of the instruments and the automatic transmission





New Steering Arm Controls. Upper View Shows Studebaker with Auxiliary Lever on Right Side, also New Front-Mounted Radius Member for Planar Suspension. Lower View Shows Buick "40" and "60" Dual-Drag-Link Construction Eliminating Previous Center Idler Lever.

control, Oldsmobile provides two wide horizontal spokes attached to the wheel rim by reversed arc members. It is standard on the 8. For the hardening of small parts the Houghton Perliton liquid carburizer is used by Gemmer, Ford, Chevrolet, and many parts manufacturers; it produces an eutectoid case up to 0.040 in. in thickness.

Equipment

The Graham headlights are built into the fenders, being wholly contoured within them. The Studebaker headlamps are faired into the apron portion of the front fenders with a lens opening simulating the radiator grille. The Oldsmobile lamps are squatted centrally on the crown, but are demountable therefrom. Dodge, DeSoto and Chrysler nestle them closely, just inside the crown. The Packard lamps are mounted directly on the aprons, eliminating the previous long neck support. Polished die-cast shields conceal the Pontiac headlamp-bracket adjusting cavities. On the 8 the parking lamps are mounted above and blended into the headlamp shell. The Cadillac "65," "75," and "90" models use a streamlined pillar with three horizontal chrome moldings. On the "60" and LaSalle a pillar with three long vertical moldings is used. This pillar effect is used on the "65," "75," and "90" to ornament the juncture of the fender apron and radiator grille. The Nash headlamp shells are mounted on the hood side panels, whereas, in the Hupmobile, they are molded into it with the continuation of the louvre chromium strips over them. Plymouth has increased the lens from 7½ to 8 in. and obtains 11¼ sq. in. more reflector surface per lamp. Oldsmobile has enlarged the lens from 6½ to 7 in. Packard, Graham, and Hupmobile place the license-plate light in the center of the trunks. The Graham tail-lamps are mounted above the belt molding, at the corner of rear quarters and trunk top. The Buick tail-light glasses come close to the fenders and are easily visible from either side of the car. The license-plate lamp is part of the trunk-lid-handle assembly in Buick and Oldsmobile. In the former it is nested in a small tubular housing above the handle shank; in the latter a large die-cast lamp housing terminates below in a boss for the shank.

Cadillac and LaSalle place the horns vertically, trumpet downward, between the grille and the front of the core. They are mounted on laminated springs to avoid transmission of vibration to the body. Bumpers are universally of the wide single-bar type and a touch of body color is often added to a decorative emblem on them.

The Oldsmobile instrument unit, made entirely of Bakelite to harmonize with the interior trim, is raised from the dash ahead of the steering wheel, with a protruding large circular speedometer dial with integral visor at the center, water temperature and ammeter indicators at the left, oil and gasoline at the right. The light switch at the left end is in the form of a large knob which rotates an illuminated dial indicating under a small rectangular opening which of the driving beams is in use. The regular upper beam pilot is still located at the top of the speedometer. Provision for installing a radio control of similar construction to the light switch is made at the right end of the unit. A centrally located roll-top compartment contains a large ash tray, provision for a lighter, and space for a package of cigarettes. Graham, who last year introduced the plastic instrument panel, provides a Tacho-speedometer. Hupmobile provides a central plastic instrument board with a vertically grilled design having the radio speaker opening at the top. The Cadillac and LaSalle plastic boards, smoothly curved and without projections, form a central raised panel with horizontal ribs accented by four wider chromium bars. The radio grille is centrally located. The starter and headlight switches are on the face of the board to the left of the steering column. The instrument numerals are white and applied to the edge-lighted cover glasses; dots are white and on dark metallic brown dials. The glove compartment lock is practically flush. A lock light for the ignition switch replaces the former map light. In the Packard senior models the heater and defroster control buttons are on the instrument panel, the pilot light for the former being in the button itself and the indicator for the latter in the bottom of the speedometer. The instruments are grouped within five circular dials. On the 6 and 8 the speedometer is located at the left with the instruments in a rectangular group at each side. An exceptionally wide glove compartment is provided. Electric clocks, to match the instruments of the senior line, have short pointers mounted on concentric rings. Pontiac has replaced the previous generator signal light with an ammeter. Packard and Chevrolet provide no numerals on their ammeters. Studebaker provides a sliding cover for the centrally located ash receiver. Buick provides ash receivers at each side of the board. Centrally recessed control buttons at the bottom of the instrument board are now found on all General Motors cars, Studebaker, Packard, Graham and Lincoln Zephyr. In the last the instrument and controls circle around the large central speedometer dial.

Narrow defroster slots built into the garnish molding now extend nearly the full width of the windshield. Mounting the heater at the center of the dash in Studebaker gives a shorter connection to the heater defroster opening and therefore a more efficient current of air. Larger vents and improvements in heaters and connecting hose have made the Plymouth and DeSoto defrosters capable of deflecting 60 per cent more warm air to the windshield.

In the Packard-6 and -8 a quickly removed panel on the back of the front seat permits insertion of a rear-compartment loudspeaker. The master toggle switch on all Packards is located under key inside the glove compartment. Hudson provides two antenna connections to permit the owner to select either a roof or a running-board antenna. The Pontiac and Oldsmobile cars with radio equipment have a Delco-Remy automatic volume-control switch to increase the volume as the car speed is increased and decrease it again for slower driving, as in city operation. The switch is actuated by the cooling-fan air stream and is mounted on the front of the cylinder-head with the switch air vane directly in the air stream. The toggle action of the tripping mechanism has a tension adjustment whereby the cut-in point of the switch can be adjusted to suit the requirements of the operation.

"Twice-a-year" chassis lubrication on the Packard-6 and -8 is obtained through the large use of rubber bushings so that there are only 17 points on the 6 and 18 points on the 8 that require lubrication. Of these, 2 need attention at 10,000-mile intervals, the remaining ones at 5,000 miles; 48 ball and roller chassis bearings are used. The 12 has 122 bearings, and there are 68 on the Super-8. Stewart-Warner has developed a reservoir cup fitted with a regular hydraulic-type Alemite fitting which, at the time of lubrication, acts as a direct inlet to the bearing surface and also allows the reservoir to fill up where the contents are retained under spring pressure.

Grilles and Sheet Metal

The new grille designs have not changed the airflow characteristics materially over the last year. Larger openings with a lessened number of grilles have helped to improve air distribution over the radiator core surface. The straighter and more widely spaced Buick grilles are separated at each side by a chromium filler ornament. The Cadillac grilles remain of the cellular type but are more massive. The V is more deeply curved, and there is a pronounced accent on the horizontal lines. The "60" grille has special raised fins at the sides and three groups of four horizontal strips continue a broken line

back along the hood. The "90" has wider and heavier bars and larger cells. The Oldsmobile-6 continues with the wide-spaced individual grilles which are now cut off at the side by the down and rearwardly sloping hood side panels. The latter effect is also found in Studebaker with its stainless-steel edged grille. The Oldsmobile-8 grille is a single die-casting with two narrow horizontal bars alternating between a wide single bar. The lower portion of the Oldsmobile grilles taper into a V formation formed by the high fender aprons. Although the 6 grille retains its rounded front, the 8 is of a modified U-shape. The base of the Studebaker is not cut off by the aprons. The Pontiac center grille retains the vertical chromium bar design extending back over the length of the hood, and the horizontal bars are less in number. The stamped grilles of all three Nash series are of identical design with horizontal bars and a die-cast crown. The leading edge of each fin is polished chrome except for a slight indentation through the center that carries the body color. The DeSoto grille is die-cast with horizontal bars that continue back along the hood to form the side louvres. A row of widely spaced louvres are located in the center of the grille. Dodge has a two-piece horizontal bar die-cast grille, separated by a central vertical design. Chevrolet's two-piece grille of horizontal design with every wide fifth bar pin-striped with vermilion enamel, has a baffle plate in the center extending to the radiator core to "straighten" the air and to prevent looking through it from the side. The grille can be removed without disturbing the sheet metal. Plymouth has a vertical stamped grille in a die-cast frame. Hudson has swung over to a horizontal bar design.

A front apron running from the fender crowns to the grille is used by all the Chrysler-built cars, cutting off the former lower portion of the grille which is no longer exposed. A chrome wing or V design is centered on the apron below the grille point. Oldsmobile's depressed, louvred aprons give a somewhat similar effect but without entirely cutting off the grille. The front portion of the Graham fenders leans forward and the same line is duplicated by the radiator grille which rises to meet the rounded hood top. The wheel opening hiding the upper part of the tires is not circular but sweeps back in a slow dropping curve. Rear fenders simulate the front ones. The Packard front fenders are tapered to a streamlined point at the rear where they join the running boards. At the front they extend below the line of the front bumper, which feature is true of most designs. The fender well openings on the senior cars have a large edge-radius and are located in relation to the rear curves so as to give the effect that the spare is fitting into a receptacle rather than being accidentally thrust therein. The high catwalks curve down at the front to blend into the radiator shell with a well-studied curvature. All Packards provide a large duct in the front fender skirt to help carry off the heated air from the engine compartment. The Cadillac front fenders now have a sharp radius at the rear upper corner. New drum-type metal tire covers are used, as well as in the case of the Oldsmobile-8, with six-wheel equipment. Oldsmobile has removable fender filler plates for engine-servicing. The Lincoln Zephyr fenders are larger, and the headlamps are more completely faired into the fronts. Combination tail-and-stop lights are recessed into the rear fenders.

The alligator-type hood is now used by LaSalle, Cadillac, Hupmobile, and Graham. The Lincoln Zephyr has a downwardly extending smooth nose blending into the V-front at the base of which are the low-positioned rectangular openings and horizontal grilles that extend to the fender crowns. DeSoto and Chrysler have changed to a hinged hood top with the side panels bolted in place, yet easily removable. For long-hood effect, all Packards have a fixed center panel as in the previous 12. Each hood-top side extends to the base of the windshield whereas the cowl ventilator lies between them and the back of the center panel. The Graham horizontal louvres are located in the hood top and are trimmed by three chromium strips encircling the entire hood. The Studebaker and Oldsmobile hoods show no louvres. In the latter, the louvre opening is concealed in the bottom of the extension of the body molding along the hood. Both upper and lower hood panels are assembled into a single unit to avoid hinge rattle or squeak. A two-piece construction with a special sturdy hinge is used on the 8 with six-wheel equipment to clear the spares. Willys has a piano-type hood hinge. Buick retains the hinged top, but the side panels are fixed.

Body

Bodies are generally lower and wider and, in a few instances, longer. Windshield openings have been made higher and wider, and visibility from all windows has been improved. Heat and sound insulation has been carried out more thoroughly. New belt-molding styles have been presented. Luggage compartments are larger and with different mountings for the spare.

The Cadillac "60," produced in only a touring-sedan model, is low-hung, the body only 65 in. high, yet without sacrifice of headroom. Graham, Hupmobile, and the Packard-6 and-8 now have all-steel bodies. The Buick "80" and "90" bodies are of steel, but with wood sills. Many makes have increased interior dimensions. In some instances this increase is due to the forward moving of the powerplant, moving the dash ahead, increasing the wheelbase, and the widening of frames and bodies. Changing the wheelbase from 122 to 125 in. and moving the engine forward, Lincoln-Zephyr has cleared up the front compartment by reducing the protrusion of the shroud over the flywheel and clutch housings. With the transmission located under the cowl and the shift rails at the right side, the shift lever has been brought up ahead of the instrument panel, by the left side of the instrument group and into a convenient position in the front compartment. Transverse toe room in the Studebaker President front compartment has been increased 8⅛ in. to 45¼ in., and both seats moved 5½ in. forward relative to the rear axle. The sedan is 65¾ in. high when loaded, and the center of gravity is approximately 26 in. from the ground. The Packard-6 and-8 have a front-seat width of 48¼ in., whereas the rear seat is 47½ in. Shoulder heights are 54⅜ and 56 in., front and rear. Headroom is increased, being 36$\frac{3}{16}$ in. front and 37⅛ in. rear. Hudson has 55 in. shoulder-height front-compartment width. Graham provides 57 in. of width front and rear, whereas the Studebaker bodies are 6 in. wider at the point of maximum width to give 55½ in. front-seat shoulder room and a rear-seat hip room of 47¼ in. Willys provides a 50-in. front seat, 47-in. rear seat with a roof height of 65¼ in. on the sedan and 64 in. on the coupe. The Chevrolet rear cushion width has been increased from 45 to 47 in.

Hupmobile and all Packards have V windshields. In the latter, the divider strip is chromium plated and a beading extends upward over the front roof radius. Both windshield halves are set in unbroken frames of soft rubber and permanently mounted. The senior cars have a 28-deg. angle. The 6 and 8 have an opening of over 13½ in., the extra vision space being obtained by slightly lowering the cowl and raising the top opening. The Cadillac "65," "75," and "90" have a 39-deg. windshield angle, the same as the previous LaSalle and Cadillac "60." Pontiac also uses 39 deg. The increased width of the Studebaker bodies permits a 6 in. greater windshield width. All Chrysler-built units have a 1-in. deeper windshield giving 12 per cent greater visibility. Buick provides dual windshield wipers driven by a single motor located

in the center of, and under, the instrument panel. The wiper arms are actuated by a system of levers and chain drive. Studebaker and Chevrolet offer as accessory equipment a windshield washer unit whereby a stream of liquid from a dash-mounted reservoir is sprayed on the windshield, after which the wiper is turned on to remove any foreign matter.

Nash provides a ventilating unit containing an air filter, motor-driven fan, and a hot-water heating element which is connected with the cowl ventilator through a stack in which the greater part of the air moisture is removed. The system maintains pressure within the body. The pivot axis on the Studebaker front-window wings is inclined.

The Buick "80" and Cadillac "65" have a slanting rear door post. For this reason the rear-quarter window slides back instead of down to approximately a 3½-in. opening. The Cadillac "60" windshield and windows are higher relative to the seat cushions to permit a better view upwards. The windshield glass is carried almost to the top of the body. The windshield pillar has been reduced by 1 in. to 2¾ in. The overall width of the center pillar also has been decreased. Structurally, the pillar remains 2 in. wide, and the decrease has been obtained by the use of narrow chrome-plated window frames. The old window-frame construction at the top has been superseded by the type used on convertibles. The Packard-6 and -8 bodies are rubber-mounted at 28 points on spool brackets with rubber washers, top and bottom. Buick utilizes 7 rubber mountings in the form of single or double large rubber shims. The points of least frame vibration were found to be on each frame rail close to the dash and just ahead of the rear motor mountings, and at the center of the rear cross-member. In addition to these three points, there are two intermediate brackets on each frame rail. Convertible bodies have hard fiber shims. Cadillac floor levels have been lowered substantially. The door-sill height of the "60" is 13⅜ in. The Chrysler-built units retain the rear rounded body contours in plan view.

The floor structural members of the Lincoln Zephyr are ½ in. deeper and of approximate box section. A convertible coupe and sedan are now available with the body units welded to the frame foundation which consists of double-box-section side-members with one channel within another, a plate welded to the edges of the upper and lower flanges, and a central X-member of similar construction. Triangular gussets are welded between the door posts and the foundation.

The Buick completely rubber-covered running boards have a wide flat band running down the center. The Chevrolet boards have an added narrow rib on top of the previous ones, with two on the outermost rib. No running boards are used on the Cadillac "60." Three parallel stainless-steel moldings finish off the lower edge of the body, and a chromium-plated guard protects the front of the rear fender.

The Graham chromium belt molding starts at the front of the radiator grille and is carried back to the trunk. The door handles are on the same line as the molding and blend into it. The Packard-6 and -8 belt molding is patterned after that previously used in the senior cars and has two pin stripes. The LaSalle stainless-steel moldings have three parallel grooves and, instead of being placed above the belt as before, are now placed below the belt curve. The Cadillac "60" has no molding, but a pronounced radius flows back from the hood along the body below the windows into the trunk, forming a natural highlight. The "65" and "75" have a two-groove molding which reaches its greatest width just back of the front-door hinge. The grooves are absent where it passes across the doors. Hupmobile utilizes a chrome belt strip terminating in the tail-lights on each side. Studebaker also has gone to the chromium belt. A similar molding surrounds the Cadillac "60" three-pane rear window and is used upon the narrow divisions.

Graham is the first production job to utilize a concealed door hinge. The Cadillac "60," Hupmobile, and Graham have both doors hinged at the front. The Studebaker sedan doors measure 34½ in. at the widest point on the front door and 26 in. for the rear. All Packard bodies have door seals of cored rubber, replacing the former sponge rubber. A very thorough study of heat-sound insulation was made by Packard who utilize eleven combinations of eight different insulating materials other than carpet, interior lining and dead-air space on the junior cars and seventeen combinations of these materials on the senior line. The materials comprise jute, printed jute, asphalt composition board, asphalt composition spray, wood composition board, 26-ply crinkled fiber, insulation fiber and rubber. Johns-Manville has developed a rock-wool filter blanket, from ¼ in. thickness up to 3 in., which can be handled without danger of breaking up the fibers. These blankets are immune to disintegration or chemical change. They are waterproof, resistant to blowtorch, and weigh approximately 3 lb. per cu. ft., and their absorption characteristics enable them to cope with all types of sound waves, particularly those of low frequency. Silence has been improved in Plymouth and DeSoto by better deadening material and a more rigid rear floor panel by pressing deeper ribs into it. Nash has developed a soundproofing material utilizing sand to add density to the surface to be damped. A rubber sealing compound is sprayed on all joints.

The Packard rear doors are operated by knobs extending up through the garnish molding, whereas the front doors retain the same remote-control handles as in 1937. The 6 and 8 have ivory-colored knobs on the window and instrument controls and the cowl ventilator. Parallel bars in chromium is the motif carried out on all hardware and instrument-panel trim. Sliding-block-door type dovetails are used by Chrysler. The inside sectors are actuated by stiff springs. The Buick "40" has a garnish molding finished in English maple with a small plastic ornament in the center of the panel below the window. The "80" and "90" moldings are of the continuous-band type with wood-grain finish on the panel below the moldings. The moldings of the "60," "80," and "90" are finished in walnut tick. A carpet-type weatherstrip has been added to the door-window garnish moldings to seal the opening between it and the glass. On the "80" and "90" this strip also has been added to the outside of the door between the glass and the bottom edge of the door window opening; in the Nash Ambassador the assist springs of the door pillars accommodate themselves automatically to varying arm lengths. Dome lights have been replaced by reading lights in the pillars which automatically switch on when the doors are opened.

The safety front seat introduced by Dodge in 1937 with the padded top edge and a cord robe-rail, has been adopted by Lincoln Zephyr and all General Motors cars. Packard is using the cord rail only. Lincoln Zephyr combines the assist cords with it and new rear-seat arm rests replace the former rail. The five-passenger Packard coupes have straight across tilting seats with very wide doors. DeSoto has a front-seat equalizer bar, keeping the seat in alignment when adjusting.

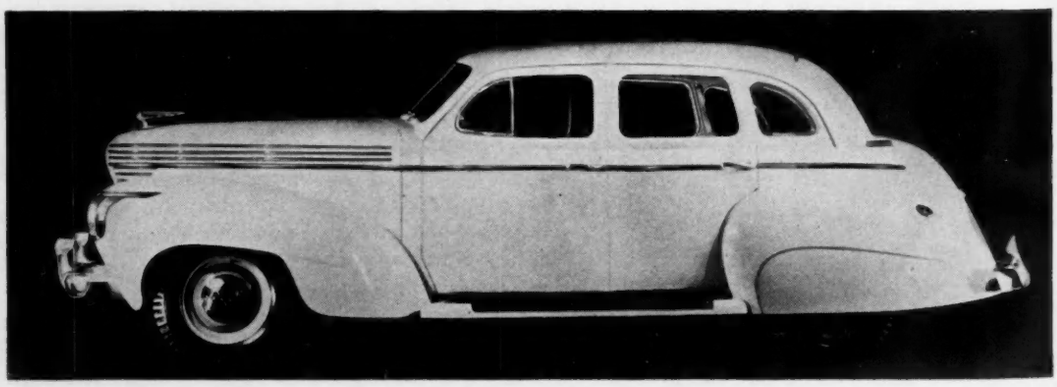
The Chevrolet deluxe is trimmed in a velvet mohair of a lighter tan shade than in 1937. A herringbone-weave broadcloth is available as optional. Seat and back cushions have three pleated ribs with four beads each. Oldsmobile is available in either a tan cloth or taupe mohair upholstery. Hudson provides a two-tone interior, and the Custom 8 has now a pleated broadcloth with a satin-finish chromium trim on the rear arm rests. Pillow-type upholstery in either mohair or broadcloth is available in the DeSoto. Upholstery on the Pontiac-6 is optional between light-tan lustrous mohair or

mixed-tan wool fabric, except in the convertibles where buffed crush-grain Spanish leather is available in six colors. The 8 comes in either mohair or heathertone wool twist. Buick uses Bedford cord, mohair fabric, and broadcloth available in tan, gray, or taupe. Plain instead of pleated upholstery is used to avoid crevices. The Packard-8 comes in a flat Bedford cord, a plain all-wool broadcloth, or in all-wool snakeskin-pattern broadcloth. The trim is of the plain tailored type with multiple stitching risers on the cushion-seat backs and on the door panels. The 12s' interiors are finished with a wide pillow-type tufted upholstery in an optional choice of ten different broadcloths. The 8 is without tufting but has a quintuple beading on both seat cushions, seat backs, and doors. Two Bedford cords and four broadcloths are available. LaSalle has two ribbed and two waffle-weave fabrics available, seat cushions having plain trim and the door center paneled by two vertical stitchings. The Cadillac "60" seat cushions are plain, and horizontal leather piping forms a bolster of the top third of each seat-cushion back. Below it are three rows of horizontal stitching forming the lower seat-cushion back into wide horizontal ribs. DuPont has developed a tough-surfaced, high-tensile-strength, sun-resisting rubber-upholstery fabric for commercial vehicles under the name of Cavalon.

Greater luggage-compartment space is obtained in the Chevrolet, Plymouth, and DeSoto by locating the spare wheel against the rear-seat panel. The Chevrolet floor is covered by a rubbed-rubber mat. The low floor and clearance for the spare required shifting of the gasoline tank, redesigning the frame rail to clear the filler, and turning the spring eyes downward. Graham provides a full-body-width trunk. The trunk-lid hinges on Studebaker and Graham are concealed, preserving a smooth exterior. Exclusive of the spare tire, the Packard-6 and -8 luggage space is 14¾ cu. ft.; Studebaker sedans, 20.1 cu. ft.; club sedans, 20.9 cu. ft.; and coupes 36 cu. ft. The largest capacity of the Buick line is found in the "81" with 18 cu. ft. on five-wheel cars and 26 cu. ft. on six-wheel. A loading capacity of 9½ and 12 cu. ft. is provided in the Oldsmobile sedan and touring sedan. The 8 with six wheels has 17¼ cu. ft. The Willys sedan and coupe have 13 and 38 cu. ft.

Heavy Vehicles

Autocar has four new light-duty trucks rated at 13,500 and 16,000 lb. gross. Models A and B have the engines under the hood, whereas the UA and UB models are C.O.E. types. The engines of the latter type are accessible by removing the right floorboard after loosening two wing nuts and the small engine hood which is held in place by two hood catches. A six-man cab has been developed for utility companies, provided with two seats and three doors. The rear seat can be folded forward so that the driver may have an unobstructed view of the winch or other apparatus. Some cabs have been fitted with a protected roof-mounted mirror to enable the operator to watch the truck-mounted crane. A fleet of 2500-gal. street-flushing trucks of streamlined design has been built. The centrifugal power is driven from a power-tower over the amidship transmission, whereas the regular transmission power-take-off drives a high-pressure hydraulic pump for snow-plow control. The American LaFrance pumper on a 200-in. wheelbase for the City of New York Fire Department carries 2000 ft. of hose, has a closed cab, and a windshield with side wings for rear-step riders. A semaphore in the cab is elevated at 30 m.p.h. to indicate the speed apart from the regular speedometer. The White "Merchandor" is a 350 cu. ft. light delivery truck with the engine entirely under the floor, accessible through trap doors and located behind the front axle. Alma has developed a front-drive axle for Ford trucks, using a cone-worm drive from a new transmission case in which the standard gears are inserted together with transfer and reduction gearing. Twin Coach has four new buses, the 31R, 35R, 40RC, and 25GU. The first has a lower floor height than the 30R and employs a horizontal drive from the transmission to the rear axle. The 35R supplants the old 37R; uses two low entrance steps instead of one large one; the power is increased 20 hp.; the engine is vertical instead of inclined; and the rear brake area is increased 50 per cent. The 40RC is similar to the 35R, except that the body has been elongated to provide space for five more passengers and supersedes the 40R. The 25GU is basically the same as the 23R except for a smaller engine and was brought out to take care of the increased demand for smaller vehicles, particularly for feeder-line purposes. The Brill H-16-SE gasoline-electric, for the first time, is able to give superior performance to the corresponding mechanical drive unit for city service. With a fuel economy not greater than 16 per cent over the mechanical, remarkable acceleration is indicated by the speed at each 10-sec. interval starting from zero with a loaded weight of 23,800 lb., 6⅛:1 axle ratio, as follows: 23, 34½, 42, 47, 51, 53½, 56, 57½ and 59 m.p.h. The Hall-Scott horizontal 180 engine with specially developed accessories is located amidship and the electric motor back of the rear axle. The Brill trolley-bus has the electric motor similarly located. It drives a double-reduction axle with the pinion shaft passing forward under the right axle shaft adjacent to the differential. The jackshaft is between the beveled pinion and the differential. The ACF H-15-S thirty-passenger coach for San Francisco-Oakland operation has service doors at both sides and front and similar emergency doors at both sides at the rear. The General Motors "740" and "739" transit-type coaches, forty- and twenty-five-passenger capacity respectively, have the loading well ahead of the front wheels with 707 and 401 cu. in. rear-mounted overhead valve engines. The "733," now twenty-one-passenger capacity, has the loading well behind the wheel and is powered by a front-mounted Chevrolet engine. Double and single deckers with the Mono-Drive transmission are found in increasing numbers in Chicago and the double decker in New York City.



Graham Styling. Note Leaning Fenders and Grille, Headlamps, Faired into Front Fenders and Door Handles in Line with Belt Molding.

Function in Modern Styling

By Frederic A. Seljé

Director, Interior Art and Body Design, Chrysler Corp.

THE purpose of this paper is to outline the relationship to, and the value of, art to the automotive industry, and its especial influence upon interior body design. To better understand our subject, let us first briefly define art:

The word "art" brings to mind a variety of work designed to please the eye and to excite the esthetic emotions. To one person it may suggest a beautiful painting; to another, a graceful statue; to yet another, a stately cathedral. Beauty is almost always associated with it; but not all things which are beautiful are works of art. All of us have been enthralled by the magnificent grandeur of a lofty mountain peak or the indescribable splendor of a flaming sunset – many have been equally enthralled by a great painting or a masterpiece of sculpture. In the one case, nature in her way has created beauty; in the other, man, by his genius, has produced beauty. The first, therefore, is not art; the second is art.

A work of art, then, is the product of a man's intellect and imagination as well as the work of his hands. According to Ruskin, "Art is the work of the whole spirit of man." The art of a people includes their sculpture, painting, architecture, and so on, and each of these branches is an art which grew out of the vital needs of the human race. For instance, primitive man needed tools with which to till the soil; dishes from which to eat and to drink; weapons for use in hunting and warfare. At first, he made the unshapen stone, a product of nature, do his work for him; then he learned how to shape it to adapt it to his needs; still later, he discovered the uses of different metals and something of the effect of fire on them; and he learned how to mold clay vessels and how to bake them in the sun or in ovens, thus acquiring a certain science.

In the course of time he began to have a sense of pride in his achievements and found satisfaction in making his tools and dishes beautiful by the addition of pleasing lines, shapes and color. Here then was the beginning of art, for he found that, when objects about him were harmonious in form, color, and arrangement, they gave him much more pleasure than when they were inharmonious; and so it has been his constant effort to achieve that harmony or beauty. In other words, man, of necessity, first had to invent certain devices if he was to live at all; once secure in this existence, he could then turn his thoughts toward beautifying these objects the better to enjoy that living. With the passing centuries, art broadened its scope and divided itself into two general classifications: the useful or mechanical arts which include weaving, embroidery, pottery making, woodworking, metal working, and so on, and the esthetic or fine arts which include painting, sculpture, architecture, poetry, music, and so forth.

It is difficult to set hard and fast rules as to the use of the preceding terms for, in some of the arts, notably architecture, beauty and usefulness are inseparable. This inseparability is also a quality of modern motor-car design because, like architecture, it too must include both beauty and usefulness. Another quality that the motor car shares with architectural structures is that they are both inclosures in which people spend much of their lives but, because one is static and the other mobile, the simile ends here.

The prime function of the automobile is, of course, highway transportation. If it is to transport persons, it must be safe, comfortable, and economical and, because it is a commodity manufactured for consumer acceptance, it should also be beautiful if it is expected to produce a profit, and it costs no more to make these vehicles beautiful than to make them ugly.

Contrary to some opinions, beauty is not obtained by the addition of applied ornaments or gadgets, but by the intelligent use of materials and processes functionally correlated to produce form, proportion, texture, and color. Surface decoration is not only costly but contrary to the rules of simplicity and directness which are the essence of good design.

In this highly competitive field, style is often the most powerful force in influencing sales, and no manufacturer can afford to indulge in the suicidal practice of leaving the styling of his product to guesswork or chance, or of burdening and restricting his designers with his own opinions, prejudices, or whims. Satisfactory results in this field can be expected only from trained and talented designers who, by experience and application, have learned how to combine the intangible qualities of art with the tangible qualities of machine-made products so that the greatest possible number of people will want to possess it. Regardless of what this service costs, the cost in unmade sales is far greater when an improperly designed product fails to register with the buying public. Most manufacturers have learned this fact either by logical reasoning or through bitter experience, and no one of them with an eye to continued profits imposes unreasonable economies upon his design staff – for here truly is a staff of business life.

Most design staffs are quite similar in make-up and function. Our own personal staff includes architects, sculptors, colorists, aeronautical engineers, mechanical and structural designers, body draftsmen, tool designers, and model makers. Theirs is the concerted responsibility of creating motor car bodies – with particular emphasis on interiors – with an atmosphere of beauty, luxury, and comfort; yet a body in which the factor of passenger safety has not been overlooked and one which can be produced within the exacting limits of costs, weight, and the working limitations of modern production machinery and materials – an accomplishment which requires a high order of constructive thinking. In order to increase its scope and efficiency, this staff must constantly project itself mentally into the future to sense style trends; explore other fields of contemporary design; seek and investigate new materials, fabrics, decorative metals, plastics, finishes, and processes; must be ever on the alert for new ideas; and must constantly broaden its horizon through research and study.

Every new design program has its own set of governing conditions which must be analyzed thoroughly and understood before a pencil can be put to paper.

Similar to general engineering practice it is the function of the department director to outline the desired procedure to his associates and to maintain close contacts every step of the way. Next, it is necessary to present the resulting material so

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graphically that associate engineers, tool designers, cost estimators, purchasing agents, manufacturing heads, service managers, sales directors and chief executives can examine and understand the proposed model in terms of their own departmental requirements and problems. Accordingly, the line drawing or the colored perspective is used only to illustrate the simplest detail. Where the problem is involved and complex, the solution is presented dramatically exactly as it will appear in actual use, that is, in form, color, and texture. In no other way can the designer adequately present his ideas and in no other way should the executive be expected to give his approval for its release to production, an act which often involves the expenditure of millions of dollars.

In our personal opinion, the interior of a car body should reflect something of its exterior styling, just as good architectural design follows a given period throughout a structure. For instance, when one approaches a colonial building, let us say, and is charmed by its lovely exterior styling and detail, one fully expects to find the same styling or period present in the interior, even to the furnishings.

Hood a Distinguishing Feature

Now the radiator enclosure and hood louvres are the principal exterior body features and, not only do they set the style note for the car itself, but they also serve to distinguish one make from another. Therefore, it is quite in order that the designer should employ these lines, or motif, around which to develop the interior scheme. With the motif thus established, he is then prepared to repeat or reflect it, with appropriate modifications, on the instrument panel, hardware, seat cushions and backs, door panels, and so on. By continuing this one motif throughout the design, harmony, repose, and a sense of fitness are created. It might be compared to a set of table silver – knives, forks, spoons – for different purposes and of different sizes and shapes; yet with one design in common.

We like to think of a motor-car body as a little room in which one spends a great deal of time and, being somewhat of a public room in that it is out of doors and exposed to view, its appointments accordingly should be somewhat formal. Its cloth lining might be either beige, gray, or neutral in color – three colors which seem to please the greatest number of persons. Any of these interiors can be made to appear gay and youthfully smart if the instrument panel and garnish moldings are finished in light colors and, inversely, dark colors in these places will give the interior an air of reserve and conservatism.

These same colors can be influenced further by the use of properly related accents of bright hues. For instance, beige takes on added charm when combined with ivory, cream, rust, or brown. Gray becomes debonair in association with blue, yellow, or vermillion. An absolute neutral is complimented by any color one may choose.

Of course such accents must be used expertly and in small areas. The escutcheon plates and knobs on the door hardware; the control buttons or other appointments on the instrument panel; the rim of the steering wheel; part of the horn button – in fact any place where molded plastic can be used to advantage is a good place for color.

We are living definitely in a color era – its influence is seen and felt everywhere, and one wonders why so little of it finds its way into our car interiors. Those of you who have traveled on such trains as the New York Central's Mercury, know how smart and charming color can make an interior and, although we are not advocating such lavish use of it in our motor-car interiors, we do wish to point out that people generally like color and respond to it subconsciously. Accordingly, we personally cannot understand why automotive sales people shy away from its use – it does not seem consistent to us that the public will accept color in everything else that it uses except in the interior of its automobiles.

We recently had the opportunity of styling a large number of car interiors for the export market wherein color was used generously. These cars were received enthusiastically by overseas dealers and customers alike, which gave us the feeling that a moderate use of bright colors in our domestic interiors would have a similar reception.

Closely related to color is pattern and texture. If a lavish design pattern is used on the door panels, a simple pattern should be used for the seat trim, and vice versa, in order to avoid overloading the design. We personally prefer not to use the same body cloth throughout the entire interior, but to use two shades of one color, placing the lighter shade on the sidewalls, door panels, and ceiling, and the darker shade on seat backs and cushions. A similar effect can be attained by using a plain cloth of a given color for the body lining and a self-patterned material of the same color for the seats. A most unusual and interesting effect can be produced if a rough material such as Kinkimo is used for the seats. Such combinations would add a much-needed freshness to car interiors without complicating assembly-line procedure or increasing costs.

In styling the interior, its function is of paramount importance because this consideration determines the form, arrangement, and appearance of the various factors. But functionalism, which should be the simplest method of arriving at serviceability and comfort, does not always produce beauty, so it is here that the designer's skill is taxed to give all possible interest to otherwise prosaic functional forms.

One of his rules is to avoid constructing ornaments for purely decorative purposes, although considerable license is allowed him in ornamenting construction.

Because he is designing for production cars, his formula is to include those features which experience has taught him will meet the needs and fancies of the greatest number of people. He knows that a simply designed and well-executed interior creates a sense of mental repose and well-being, and that well-upholstered seats with deep cushions of proper height and contour produce a physical comfort which eliminates fatigue and adds to the pleasure of riding. He knows also that, in addition to luxurious and spacious seats, it is important that jolting, vibrations, noise, drafts, dust, water and undue heat and cold be eliminated if the occupant's comfort is to be complete.

But perhaps the most important function of the motor-car body is to provide the greatest possible safety for its passengers. And here the all-steel shell and shatterproof glass are of great protective value. The interior of the body should be made safe by the elimination of all possible projections with which a passenger may collide or over which he might trip, and all surfaces which come into normal contact with his person should be as soft or smooth as construction will permit. Door handles and window regulators need not project unduly; knobs, switches, and controls can be flushed to the surface of the instrument board as already has been demonstrated. The importance of incorporating every safety measure possible into interiors cannot be too strongly stressed; the best we can provide is far from adequate. And now in closing:

What about the future of automotive design and styling? The markets are full of new and intriguing materials – scores of interesting processes and finishes have been developed – the consuming public already has accepted eagerly the modern order of design and, if the industry avails itself fully of these advantages and factors and gives the designer a free hand, then the future is full of promise, surprises, and rewards. Beyond that, there is little that anyone can predict.

Hypoid Gears, Axles, and Lubricants

By W. A. Witham
Gleason Works

THIS paper discusses the most recent developments and best practice in both the design and manufacture of hypoid gears and axles.

Due to improved machines, methods, and materials, hypoid gears now are being produced to higher standards of accuracy and precision than has heretofore been possible to produce any type of rear-axle-drive gears.

To utilize such improved gears to the greatest advantage, similar advances are being made in the design and manufacture of hypoid axles. Greater rigidity of the gear mountings and the highest possible degree of precision in producing uniform axles are required, as well as proper provision for control of the tooth bearing in assembling the axle.

Latest methods for producing hypoid gears are discussed, including an analysis of the Formate type of gear. A summary of the results of a number of deflection tests on various types of axles serves to illustrate various factors entering into axle design.

THE idea of a pair of tapered gears, in which the axis of one member is offset from the axis of the other, is not of recent conception.

Prior to 1925, however, no method was known for correctly designing or cutting the teeth of such gears. Practically all attempts to cut a pair of bevel gears for operation with offset axes were confined to cases where one or both members had skew teeth and both members were generated as bevel gears. Such gears do not have correct profile contact when meshed in offset position. Accordingly, the tooth action is rough, and the gears are noisy.

In 1925, the Gleason Works developed a new method for producing hypoid gears for automobile rear-axle drives wherein the gear was cut the same as a spiral-bevel gear but the pinion was generated in an offset position corresponding to the position in which it was to run in the axle. A great deal of mathematical analysis was necessary to derive formulas for the design of the teeth and the determination of machine settings for correct generation. A completely new pinion-generating machine was designed. This machine had as its principal feature a vertically adjustable workhead to obtain the hypoid offset. In addition, angular adjustments of the cutter were provided which made possible complete control of the tooth-bearing. The same methods of control were applied later to the generation of spiral-bevel pinions to obtain greater refinements than previously had been possible.

In the production of hypoid gears no additional operations are necessary or any changes in the general methods used in spiral-bevel gear production.

It was found immediately that these hypoid gears were inherently quieter and stronger than spiral-bevel gears. The Packard Motor Car Co. adopted hypoids for production in 1926 and have used them exclusively since. Several other car manufacturers adopted hypoids in the following eight years and, during the past two or three years, the change to hypoid gears for rear-axle drives has been rapid.

It appears that hypoid gears were developed and introduced before their time – before the need of quieter, stronger axles had been felt seriously.

With the ever-increasing motor torques and the continued demand for quieter performance, the eventual use of hypoid gears was inevitable. Other inherent advantages have served only to hasten their almost universal adoption.

The practical limit of lowering car heights without serious loss of headroom had been reached with spiral-bevel rear axles. The possibility of further decreasing height of the car by the full amount of the hypoid offset with no loss of headroom was most attractive in view of the public demand for lower cars.

Early prejudices against hypoid gears were formed because they required special extreme-pressure lubricants for satisfactory operation. This feeling probably was justified in that there were very few satisfactory hypoid lubricants in 1926. Due, however, to the cooperative efforts of a number of oil companies, the early users of hypoid gears, and the Gleason Works, a considerable number of highly satisfactory hypoid lubricants have been developed and are now being marketed on a sufficiently large scale to insure proper service to every automobile owner.

The increasing use of extreme-pressure lubricants in spiral-bevel axles because of the increasingly heavy loads imposed, also has caused many prejudices to be discarded.

The fact that hypoid gears are inherently quieter than spiral bevels and, accordingly, easier to produce to high standards of quietness, also has contributed largely to their rapid adoption.

Due to constant improvements in methods and machines, hypoid gears are now being produced to a higher degree of accuracy and precision than it has heretofore been possible to produce any type of rear-axle-drive gears.

The introduction of the single-cycle method of cutting Formate or non-generated gears has been perhaps the most noteworthy advance in this direction.

Formate gears differ from generated gears primarily in the

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amount of profile curvature. In a generated set, correct conjugate tooth action is obtained by making the profiles of both the gear and pinion substantially involute. (See Fig. 1.) The Formate gear profile is made straight, being formed by a cutter having straight blades. The pinion is generated to be correctly conjugate to this type of gear. The actual difference in profile shape between a generated gear and Formate gear of say 10-in. diameter and 40 deg. spiral angle, is only 0.003 in. to 0.005 in.

Now, the Formate gear is displacing all others for quantity production chiefly because of a new machine and a new type of cutter which will produce more accurate gears at a faster rate than any bevel gear could be produced formerly.

The cut is produced in a new and original manner, the feed being imparted to the cutter by setting successive blades radially beyond the preceding blades so that each removes about 0.001 in. of stock. A feeding mechanism is thus eliminated. One tooth is finished with each revolution of the

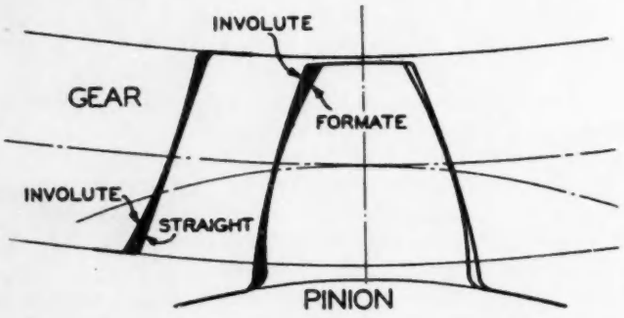


Fig. 1 – Profile Curvature of Formate and Involute Gears

cutter. Indexing is accomplished while a gap between the last finishing blade and the first roughing blade is opposite the gear face, without withdrawing either the cutter or the work. Thus, while the tooth is being cut, the only motion in the machine is the rotation of the cutter.

The greater rigidity of this machine, in combination with the single-cycle method of cutting, enables us to produce gears to an extremely high degree of precision, and having an excellent surface finish.

The teeth of Formate gears also are being ground in quantity production with highly satisfactory results. This method of finishing has the advantage of eliminating the effects of distortion due to hardening, as well as of producing an excellent finish and extremely accurate tooth spacing.

The hypoid pinion is finish-cut in a generating machine, in an offset position corresponding to its offset in the axle. The cutter represents a tooth side of the hypoid gear, and the same relative motion is imparted between the cutter and the pinion to be cut as exists between the finished gear and pinion. With the adjustments provided on this machine, tooth surfaces can be produced that match those of the mating gear along their entire length and across the whole depth of their profiles. However, for adjustability, that is, to provide a certain range of running positions allowing for a non-rigid axle mounting, the bearing area is somewhat reduced. Any desired deviation from a full conjugate bearing can be obtained lengthwise or on the profile.

Thus, in the generation of the pinion, we have complete control of all of the tooth bearing characteristics, including length, width, and general shape of the contact area. The most desirable characteristics of the tooth bearing are obtained by determining axle deflections and observing the movement of the tooth bearing under appropriate loads. These data are then correlated with results obtained by running the gears in a rigid testing machine and are applied to the development of the finish-cut pinion.

Lapping

Lapping is now used universally as a surface-finishing operation after hardening to produce a degree of smoothness that would come only after many miles of normal operation of the gears in the axle.

Due to the sliding action present at all points on the tooth surfaces of hypoid gears, the lapping action is sufficiently uniform over the entire tooth surfaces so that no change in the true generated tooth form is encountered.

The lapping action is faster in hypoid gears than in spiral bevels due to the increased sliding action. This action also helps to carry the lapping compound, which is suspended in a vehicle of fluid oil between the tooth surfaces.

The lapping machine is so designed as to traverse automatically the localized tooth bearing from the toe to the heel of the teeth while lapping, thus smoothing the entire tooth surface.

Gear Design

With regard to the design of hypoid gear sets, there are a number of standards and general-practice rules that have been evolved during the years that we have been producing hypoids.

Due to the offset position of the pinion, the profiles of the opposite sides of the teeth are not symmetrical. In the usual arrangement the concave side of a hypoid pinion has a flatter profile than the convex side. In order to maintain approximately the same conditions of tooth contact on the two sides the pressure angles are unequal. By equal conditions of tooth contact we mean equal arcs of action and duration of contact, similar relative radii of curvature of the tooth profiles, and equal freedom from undercut on the two sides. Pressure angles of approximately 17½ deg. on the drive side and 25 deg. on the coast side are most satisfactory for general use.

With regard to the determination of the spiral angle of hypoid gears, a number of factors have been considered.

With an increase of spiral angle we have a greater number of teeth in contact, and an increase in the pinion diameter. On the other hand, we also get an increase in the axial thrust and in the normal tooth load. We have found that the load-carrying capacity or surface durability increases faster than the axial thrust. The best balance of all of these factors appears to be gained with a pinion spiral angle of 50 deg. For the average automotive hypoid this value gives a spiral angle for the gear of roughly 25 deg., which is well below that for the average spiral-bevel passenger-car drive.

In determining the correct proportions of the blanks for hypoid gears, several interdependent factors must be considered, such as the ratio, spiral angles, pitch angles, cutter radius, and offset. These factors are so interrelated that a change in any one of them will affect all the others. In spiral-bevel gears the ratio of the number of teeth in the gear to the number of teeth in the pinion determines the pitch angles, such factors as spiral angle and cutter diameter being independently established.

Tooth Loads

The total normal tooth load on either spiral-bevel or hypoid gears is calculated by dividing the tangential load (which may be determined from the known motor torque) by the cosine of the spiral angle of the gear and the cosine of the normal pressure angle.

In the hypoid set, due to the unsymmetrical relationship between the gear and pinion, a pinion spiral angle of 50 deg. and 1¾-in. offset would require a gear spiral angle of say

25 deg. A set of spiral bevels comparable for smoothness of action would require a spiral angle of at least 45 deg. on both gear and pinion. We can assume a 16-deg. pressure angle for both. The normal tooth load for the hypoid gear is then 15 per cent greater than the tangential load and, for the spiral bevel gear, 47 per cent greater than the same tangential load.

Thus, due to the lower spiral angle of the hypoid gear, the normal tooth load is substantially less than on a spiral-bevel gear of the same diameter and carrying the same torque.

The limits for tooth loads recommended by the Gleason Works for automotive rear-axle drives are 1600 lb. per in. of gear face width in direct drive, or 4200 lb. per in. of face through low gear.

Hypoid gears are particularly well suited for use in high-reduction truck axles where the increased diameter of a hypoid pinion over a spiral-bevel pinion gives a very considerable increase in tooth strength. For such applications, the hypoid offset should not be greater than one-eighth of the gear diameter.

Surface stress, or surface durability, of hypoid gears is calculated on the same basis as for spiral-bevel gears.

Gear Mountings

The maximum potential load-carrying capacity of automobile rear-axle drive gears is never realized due chiefly to flexibility of the mountings, which allows concentration of the tooth bearing with corresponding concentration of stress.

The degree of rigidity of the gear mountings determines the length of the tooth-bearing allowable. The more rigid the mountings, the greater the length of tooth contact that can be used and the lower the resulting surface stress. Excessively flexible mountings require a short-tooth bearing located close to the toe under light loads in order to prevent excessive concentration of load at the large end under heavy loads.

It is not possible to predict with any degree of accuracy the behavior of gears in an axle from a visual or mathematical study of the axle. The only satisfactory method is to study the actual movement of tooth bearing under loads progressing from zero to the maximum that can be imposed in service, to amplify such a study with actual measurements of the deflections of the various members of the axle, and to correlate such data with displacement checks in a rigid testing machine. By such a method it is possible to develop a tooth bearing of such a shape and size as will maintain a satisfactory contact under all loading conditions.

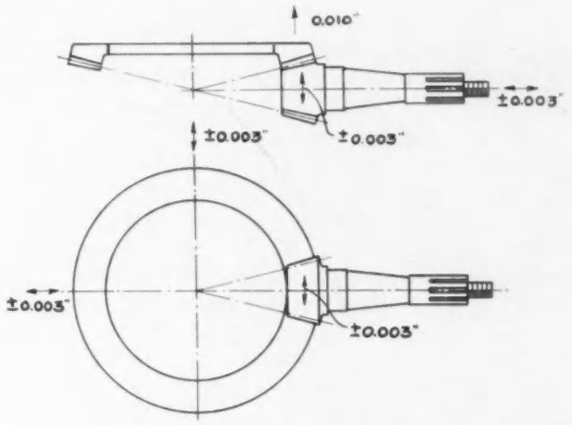


Fig. 3 – Maximum Permissible Displacements

Axle deflection studies are made in a testing machine designed expressly for this purpose (see Fig. 2). The axle is motor-driven through a gear reduction unit to give about 6 r.p.m. of the pinion. Load is applied by means of a Prony brake of ample size to eliminate variations in load. Indicators are mounted from a rigid bracket so as to measure deflections of both gear and pinion along three axes. If any of these deflections proves excessive, an analysis is made of various contributing factors to determine how the unit may best be stiffened.

Tooth bearing is observed by painting the gear with a mixture of powdered red lead and oil after each successive load is applied.

By correlating the data obtained in these tests with service reports and other laboratory tests over a period of years, we have determined satisfactory limits for deflections of the gear and pinion mounted in the axle and subjected to a load equivalent to the maximum load which can be applied when the axle is in actual service. This load is usually maximum motor torque through low gear.

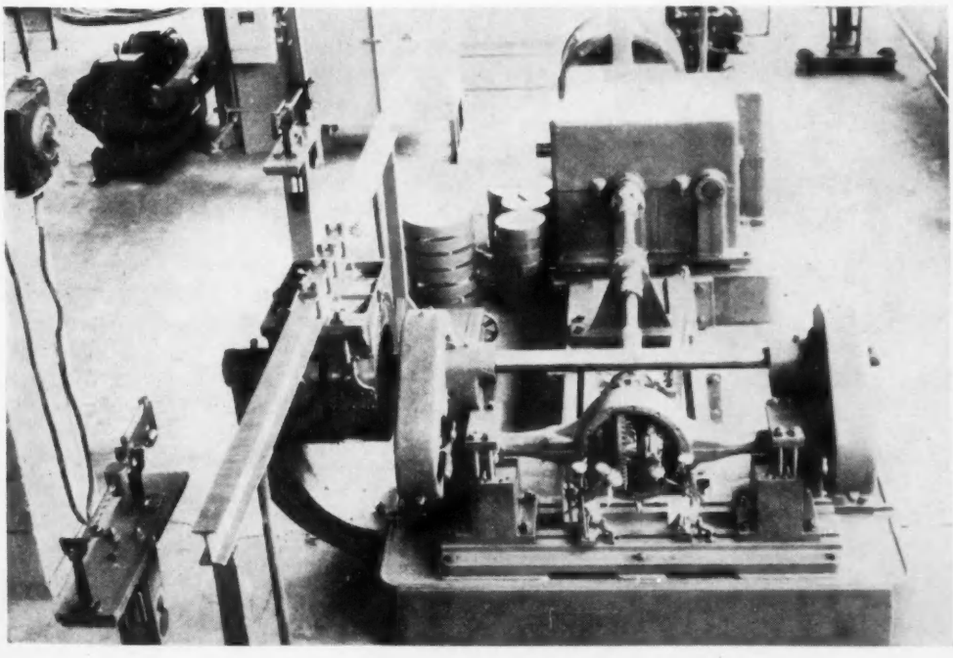


Fig. 2 – Testing Machine for Axle-Deflection Studies

Maximum displacement of the pinion should not be greater than 0.003 in. in any direction from its central position, axially, vertically or transversely as shown in Fig. 3.

Maximum displacement of the gear vertically, or in the direction of the pinion axis, should not exceed 0.003 in. Movement of the gear away from the pinion at the mesh point should not exceed 0.010 in.

In order to hold the gears within these limits, the bearings must be of ample size, must be supported properly, and the gear and pinion mountings must be tied together rigidly to prevent misalignment.

Elastic yield in the bearings is a function of the size and internal proportions. Bearings usually are selected by consid-

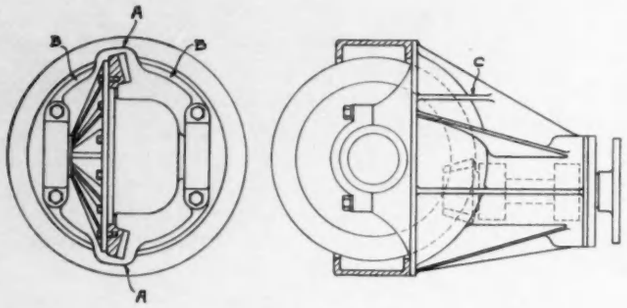


Fig. 4 – Conventional Carriers Showing Arrangement of Ribbing

eration of cost and size and, thus, often are undersized with respect to their internal rigidity. Internal yield can be reduced, however, in most types of bearings by preloading, a practice which has been proved to possess great merit. In the case of the double-row rigid-type bearing, the amount of preload must be built into the bearing by the manufacturer. In this case, the most desirable amount should be determined by testing the complete axle, and then proper inspection methods devised to insure uniformly preloaded bearings in production. With tapered-roller bearings, or with certain types of single-row ball bearings used in pairs, the preload must be controlled by adjustment at assembly.

Straight roller bearings, used at the head of the pinion, or in the outboard support of straddle-mounted pinions, are not preloaded for various reasons. To insure proper rigidity they must, therefore, be selected carefully for a radial freedom of not over 0.0004 in. and must be of ample capacity to carry the imposed loads without excessive deflections.

Differential side bearings, either ball or tapered roller, must be preloaded to keep deflections within proper limits. General practice is to apply sufficient load to increase the distance across the back end of the pedestals by roughly 0.015 in. Various types of axles, of course, require different treatment.

The main point is that preloading has been necessary in every axle that we have tested to keep the displacements of gear and pinion within proper limits. The amount of preload required varies with various types of bearings and methods for supporting the bearings – but it can be determined by measuring deflections of the gears under appropriate loads. Further tests for determining the operating characteristics of the bearings under such preloads should be made as considerations of rigidity show them to be necessary.

Rigidity of the bearing supports must be secured by the addition of ribbing to the body of the carrier. Proper location of ribs to tie in the bearings most effectively is gained as a result of careful study and design rather than by the lavish use of iron. A few types of ribbing that we have found to be effective in adding to the rigidity of the gear mountings are shown in the sketches of conventional carriers illustrated in Fig. 4.

Vertical ribs of as large proportions as space will permit are desirable in providing resistance to vertical pinion displacements. The cutaway portion at *A* in Fig. 4 should be reinforced to prevent excessive distortion of the back flange.

Gusset ribs, *B*, extending in toward the gear as closely as possible are effective in distributing the loads between the two pedestals. The horizontal rib shown at *C* is often used to improve rigidity of the pedestals and carrier flange. By offsetting the gears transversely with relation to the housing, as shown in Fig. 5, better distribution of the load between the two pedestals is gained as well as adequate space for proper gear backing and a differential case flange of ample thickness and proper proportions to resist bending.

I have discussed the most common type of axle carrier with respect to rigidity but it will be found, in general, that the same remarks will apply to other types.

Gear-Tooth Breakage

Failure of gear teeth in service by fracture has been found in practically every case to be due to fatigue, usually of the pinion teeth. We have, in the past 10 or 15 years, run a great number of tests to determine the fatigue life of spiral-bevel and hypoid gears subjected to maximum motor torque through low gear. No attempt to simulate actual road conditions has been made other than to run the gears mounted in the same axle assembly as occurs under the car. By correlating the results of these tests with the service records of various manufacturers, we have reached some definite conclusions as to the fatigue life required of a good axle.

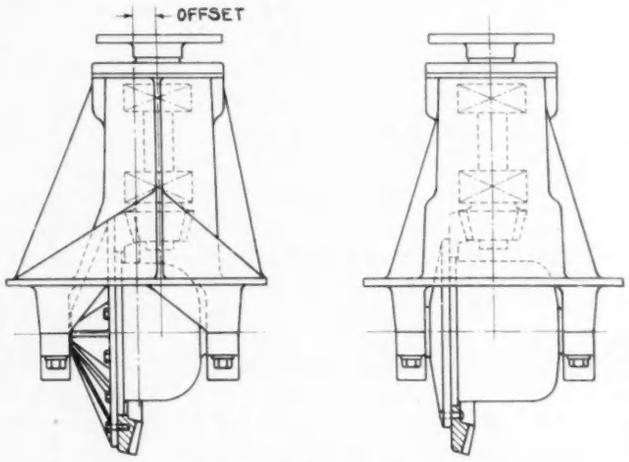


Fig. 5 – Carrier with Gears Offset Transversely with Relation to the Housing

A spiral-bevel axle that will withstand 200,000 cycles of the pinion under maximum motor torque through low gear in the laboratory tests has been found to give consistently good service in the field. Axles having shorter fatigue life in the laboratory test have shown relatively less durability in actual service.

Hypoid gears have given consistently many times the fatigue life of the best spiral bevels when tested under the same conditions. The average of some 40 tests on hypoid axles that we have run to date is 1,135,000 cycles of the pinion for a fatigue failure. The minimum figure for a failure of a hypoid axle was 287,642 cycles in a test on an experimental axle. Even this figure is considerably higher than for most spiral bevels. The maximum has not been determined. One set of axles was run under maximum low-gear torque for 4,860,000 cycles without failing.

The wide variation in fatigue life of gears of similar general design and stressed nearly alike has enabled us to determine to some degree the relative importance of various factors affecting fatigue strength.

Rigidity of the gear mountings probably has the greatest effect of any one factor on fatigue life of rear-axle gears. In practically all cases of gear-tooth breakage, a fatigue fracture starts at the root of the pinion tooth at the large end where the stress is concentrated under heavy loads. Reducing stress concentration at the heel by the use of a short toe bearing in assembly is limited by the increasing difficulty of obtaining quiet gears as the contact is shortened and positioned closer to the toe. A much more effective improvement can be made by stiffening the gear mountings to reduce deflections of the gear and pinion.

Pinions are designed with a generous fillet at the root, the cutters being made with as large a radius as practical.

Improvements in tooth spacing are reflected in lower tooth stresses and greater durability of the gears.

It is doubtful whether we will ever have a case of tooth breakage in hypoid axles in ordinary service, providing that reasonable care is taken in the design of the axles, in the cutting of the gears, and in the final assembly.

E.P. Lubricants and Oil Tester

The testing of hypoid lubricants for load-carrying capacity has been carried on at the Gleason Works for the past 12 years in the machine shown in Fig. 6, which was designed specifically for this purpose. The performance of the oil in a running test in standard production hypoid axles is used as the criterion of its suitability for automotive hypoid rear-axle lubrication, in so far as its load-carrying capacity is concerned. The usual specifications of cold pour, viscosity, and so on, are not investigated in any detail, as these specifications are commonly settled upon by the lubricant and axle manufacturers.

The machine shown is in the form of a "four-square" axle-testing machine, using only two axles, however, to form a closed system. A torque load is applied by means of the "wind-up" mechanism enclosed in the box at the right and the system rotated by means of a motor concealed behind the post at the left.

Two standard production hypoid axles of a model that has been proved satisfactory with regard to gear deflections are used for the test. They are inspected carefully for correct positioning of the tooth-bearing and factory bearing adjustments.

The machine is first brought up to a speed equivalent to approximately 25 m.p.h. in the car, and then the load is applied gradually, building up to the equivalent of full motor torque through second gear in about 1½ min.

The test is run with the load on the drive side of the gears until the temperature reaches 200 deg. fahr., and on the coast side for 8 hr., holding the temperature of the oil in the axle to a maximum of 250 deg. fahr. by water-cooling.

The results obtained in these tests have been correlated with findings from road tests and with service records. It has been found to be true in practically every instance that lubricants tested in the laboratory under the conditions described show results in actual service that are similar to a remarkable degree. For example, some lubricants have been what might be termed borderline cases in laboratory tests, the gears showing only fine hair-like "action lines," hardly serious enough to be called scoring, but indicating a mild distress of the bearing surface which eventually would result in excessive wear. In actual service, such lubricants have shown the same characteristics in substantially the same degree. In no instance has a lubricant that has failed definitely in the laboratory test been found suitable for lubrication of hypoid rear-axle gears in actual service.

Many attempts have been made to correlate the findings of these tests with various simpler tests but, to date, the results have been disappointing.

It may be observed that the problem of the lubricant manufacturer in providing a lubricant of sufficient E.P. value has been simplified to a considerable extent by comparatively recent developments. Greater accuracy in producing the gears, improved surface finish, the design of more rigid carriers, and superior control of the tooth bearing shape, all serve to increase the carrying capacity of the gears and tend to decrease the necessary E.P. value of the lubricant.

A few of the primary requirements of an E.P. oil suitable for hypoid rear axles are: first, sufficient load-carrying capacity or E.P. value; second, absence of abrasive action; third, stability, both in service and in storage; and fourth, absence of corrosive action.

In the design of a hypoid axle it is essential that provision be made for accurately positioning the gears at assembly to give the desired tooth bearing. This positioning is accomplished usually by means of shim or screw adjustments. A very desirable trend is toward designs in which the original factory adjustments cannot be changed easily.

An opening must be provided in the axle housing, usually in the back coverplate, so that an inspection of the tooth contact can be made after the axle is assembled completely.

For a high degree of surface durability in hypoid gears it is necessary to maintain the maximum possible degree of surface hardness in both the gear and pinion. Because the usual type of penetration test for hardness does not reveal a soft skin, we have found that the best check on surface hardness is a file test. It should be impossible to "cut" the tooth surface of either member with a sharp XF file.

Of course, the use of hypoid gears is not limited to the automobile rear axle. For any arrangement of machinery where it is desired to have the gear and pinion shaft cross each other, and where the gear ratios range from 1:1 up to 6:1, hypoid gears are the logical solution. Generating machines are available for cutting hypoid gears over 5 ft. in diameter, the greatest pinion offset possible at present being roughly 9 in.

Hypoid gears are being used in locomotive and high-speed electric streetcar drives and in various types of industrial applications such as paper-mill, steel-mill, and wire-drawing machinery.

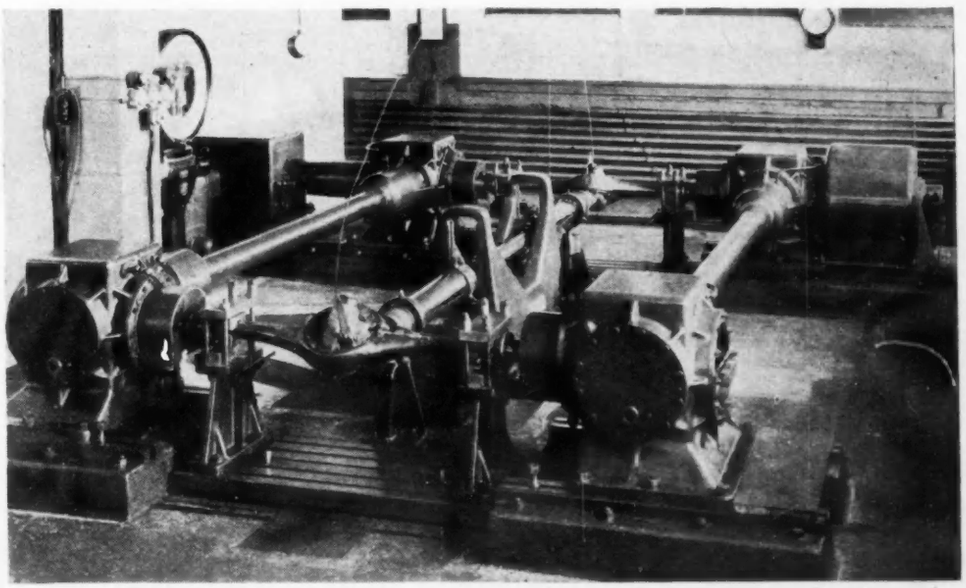


Fig. 6 – Machine for Testing Hypoid Lubricants for Load-Carrying Capacity

Variables Affecting Flame Speed in the Otto-Cycle Engine

By C. L. Bouchard, C. Fayette Taylor, and E. S. Taylor *
Massachusetts Institute of Technology

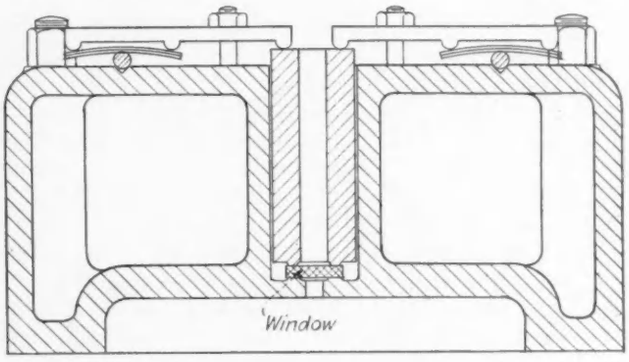
IN the investigations reported in this paper flame-trace photographs were taken on a moving film through a glass-window slot in an engine cylinder to show the effects of various operating conditions on the rate of flame travel across the combustion-chamber. The tests were made with a small L-head single-cylinder engine in the Sloan Automotive Laboratories at the Massachusetts Institute of Technology. The technique is similar to that used by Withrow and Boyd in flame studies reported in 1931.

This investigation covers a considerable range of operating conditions, including altitude, with and without supercharging, inlet temperature, humidity of the intake air, engine speed, ignition timing, and fuel-air ratio.

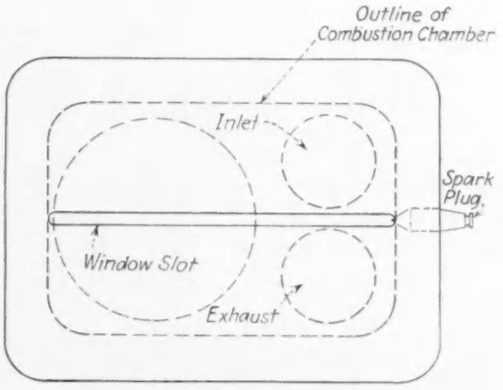
In general, the results show that flame speed decreases with increasing altitude in an unsupercharged engine. Either supercharging or reducing the exhaust pressure with inlet pressure constant, tends to increase flame speed. The flame speed decreases with increasing inlet temperature and with increasing humidity. Observations of the effects of revolutions per minute, fuel-air ratio, and spark-advance confirm the results of previous investigations.

THE rate of flame travel in a spark-ignition engine cylinder is the chief factor controlling the rate of pressure rise, and hence it has an important influence on engine performance. Variations in the rate of flame travel (referred to the speed of crank rotation) are not usually sufficient to cause large variations in output and efficiency, provided optimum spark timing and normal fuel-air ratios are used. However, with modified spark timing or with lean mixtures, variations in flame speed may exert a marked influence on output and efficiency. The time of combustion determines the piston position at which maximum pressure occurs and, therefore, has a marked influence on the tendency to detonate. Flame speed, through its influence on rate of pressure rise, also will have a direct effect on the detonation tendency.[1]

A large number of investigations have been made on the nature of flame movement in closed vessels of various kinds,[2-13 inc.] consisting of constant-pressure bombs,[2-9 inc.] and constant-volume bombs.[10, 11, 12, 13] With very few exceptions, the main purpose of these investigations was to study the movement of the flame front on the assumption that results obtained by means of the explosion vessel could be interpreted



Cross Section of Head
Showing Window and Method of Clamping



Sketch of Combustion Chamber

Fig. 1 – Cylinder-Head and Window

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so as to yield useful information with regard to flame movement in the Otto-cycle engine.

More recently, studies have been made of flame development in spark-ignition engines through the use of transparent windows of various kinds in the cylinder-head. Interesting work of this kind has been done by Withrow et al.,[14, 15] Marvin et al.,[16, 17] and Schnauffer.[18] The method used in the present research, namely, that of photographing the progress of the flame front through a narrow slot extending across the combustion-chamber was first used by Withrow and Boyd.[14]

The object of the work reported herewith was partly to confirm the work of earlier investigators, but chiefly to investigate the effect on flame development of operating variables which had not been investigated in previous work. Chief among these are the effects of increasing the altitude

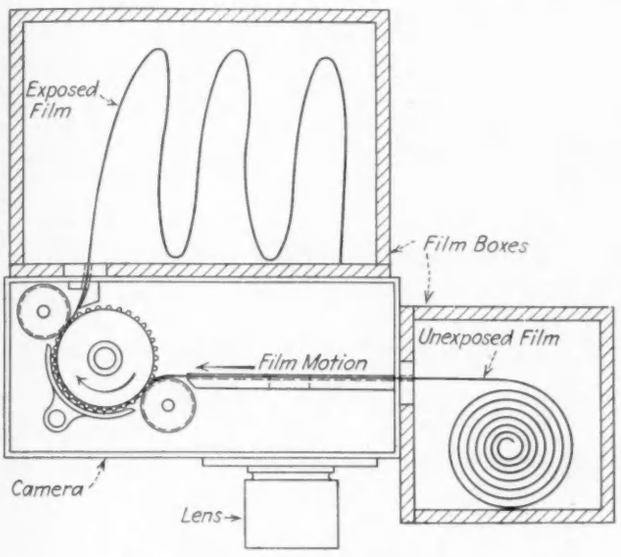

Fig. 2 – Camera and Film Boxes

both with and without supercharging, and the effects of variations in inlet temperature and humidity.

Apparatus

The engine used for this investigation was a one-cylinder L-head marine motor with a 4⅜-in. bore and 4½-in. stroke. It was equipped with a special cylinder-head having a narrow window extending from one end of the combustion-chamber to the other. This window, of Pyrex glass, was held against a very accurately machined surface in the head by a metal frame fitting down into the opening, as is shown in Fig. 1.

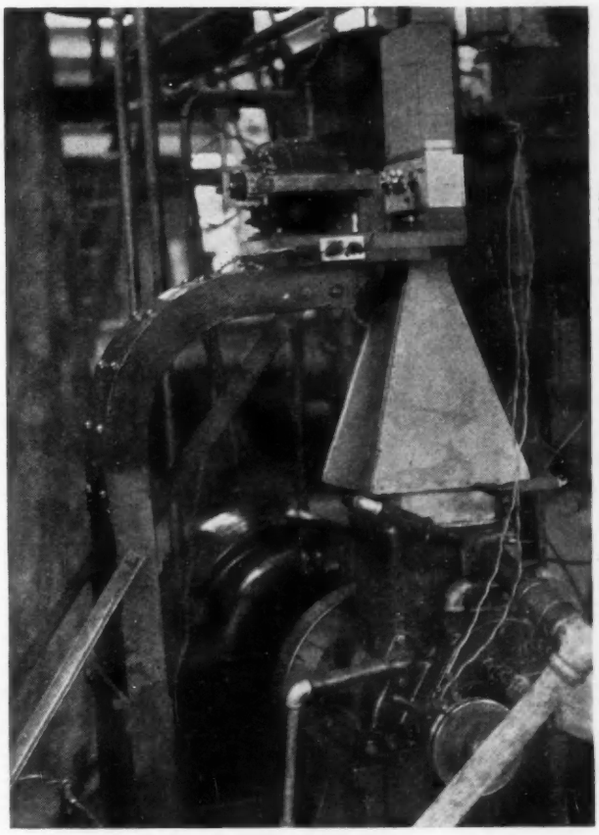

Fig. 3 – Camera Mounted Above Engine

The novel feature of this window-mounting is that no gasket or cement is required to seal the window in order to prevent the high-pressure gases from escaping. Another feature is the ease and rapidity with which the window can be removed to be cleaned. The total time for removal, cleaning, and replacing is less than 5 min.

The camera, shown in Fig. 2, utilized standard 35-mm. motion-picture film driven at uniform velocity by a synchronous motor. It was mounted so that the film motion was at right angles to the axis of the window. It contained no shutter, the film being exposed continuously to the window in the head of the motor. Only the light emanating from the cylinder due to combustion reached the film. In Fig. 3 the camera may be seen mounted directly above the head of the motor.

The engine was fitted with an ignition system comprising a commutator on the crankshaft operating a special relay trip

[1] See N.A.C.A. Technical Note No. 452, March, 1933: "The Importance of Auto-Ignition Lag in Knocking," by E. S. Taylor.
[2] See N.A.C.A. Technical Report No. 176, 1923: "A Constant-Pressure Bomb," by F. W. Stevens.
[3] See N.A.C.A. Technical Report No. 280, 1927: "The Gaseous Explosive Reaction – The Effect of Inert Gases," by F. W. Stevens.
[4] See N.A.C.A. Technical Report No. 305, 1928: "The Gaseous Explosive Reaction – A Study of the Kinetics of Composite Fuels," by F. W. Stevens.
[5] See N.A.C.A. Technical Report No. 337, 1930: "The Gaseous Explosive Reaction at Constant Pressure, The Reaction Order, and Reaction Rate," by F. W. Stevens.
[6] See N.A.C.A. Technical Report No. 372, 1931: "The Gaseous Explosive Reaction – The Effect of Pressure on the Rate of Propagation of the Reaction Zone," by F. W. Stevens.
[7] See N.A.C.A. Technical Report No. 531, 1935: "The Effect of Water Vapor on Flame Velocity in Equivalent CO-O_2 Mixtures," by E. F. Fiock and H. K. King.
[8] See N.A.C.A. Technical Report No. 532, 1935: "The Soap-Bubble Method of Studying the Combustion of Mixtures of CO and O_2," by E. F. Fiock and C. H. Roeder.
[9] See N.A.C.A. Technical Report N. 553, 1936: "Some Effects of Argon and Helium upon Explosions of Carbon Monoxide and Oxygen," by E. F. Fiock and C. H. Roeder.
[10] See S.A.E. TRANSACTIONS, 1921, Part I, Vol. 16, pp. 465-509: "The Nature of the Flame Movement in a Closed Cylinder," by C. A. Woodbury, H. A. Lewis, and A. T. Canby.
[11] See N.A.C.A. Technical Memos Nos. 547 and 548, 1930: "Contribution to the Study of Normal Burning in Gaseous Carbureted Mixtures," Parts I and II, respectively, by M. R. Duchene.
[12] See N.A.C.A. Technical Memo No. 694, 1932: "Combustion of Gaseous Mixtures," by M. R. Duchene.
[13] See *Journal of Chemical Physics*, May, 1934, Vol. 2, p. 283: "Determination of the Speed of Flames and the Temperature Distribution in a Spherical Bomb from Time-Pressure Explosion Records," by B. Lewis and G. von Elbe.
[14] See *Industrial and Engineering Chemistry*, May, 1931, Vol. 23, pp. 539-547: "Photographic Flame Studies in the Gasoline Engine," by L. Withrow and T. A. Boyd.
[15] See S.A.E. TRANSACTIONS, August, 1936, pp. 297-303, 312: "Slow Motion Shows Knocking and Non-Knocking Explosions," by L. Withrow and G. M. Rassweiler.
[16] See N.A.C.A Technical Report No. 399, 1931: "Flame Movement and Pressure Development in an Engine Cylinder." by C. F. Marvin, Jr., and R. D. Best.
[17] See N.A.C.A. Technical Report No. 556, 1936: "Further Studies of Flame Movement and Pressure Development in an Engine Cylinder," by C. F. Marvin, Jr., A. Wharton, and C. H. Roeder.
[18] See S.A.E. TRANSACTIONS, January, 1934: "Engine Cylinder Flame Propagation Studied by New Methods," by K. Schnauffer.

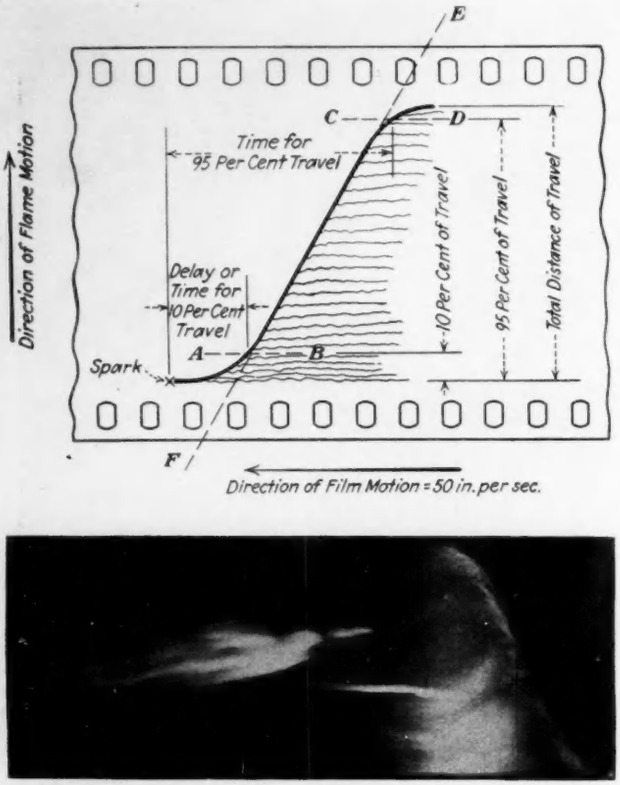

Fig. 4 – Flame Trace (*below*) and Curve of Flame Speed Against Time (*above*)

circuit employing a Thyratron tube. This arrangement provided much less variation in spark timing than the usual "breaker" mechanism. This accuracy was important since all the film measurements were made from the spark as an origin.

Explanation of Flame Photographs

Fig. 4 indicates how a trace of the movement of the flame front appears on the film. The spark is visible through the window and is recorded on the film.

Photographs of the entire flame front, published by Withrow and Rassweiller[15] show that it is of irregular shape and that it does not generally remain at right angles to a slot such as that used in these tests. The present method, however, will indicate trends in the average velocity of the front, provided that there is no systematic change of flame-front shape with the variable being investigated.

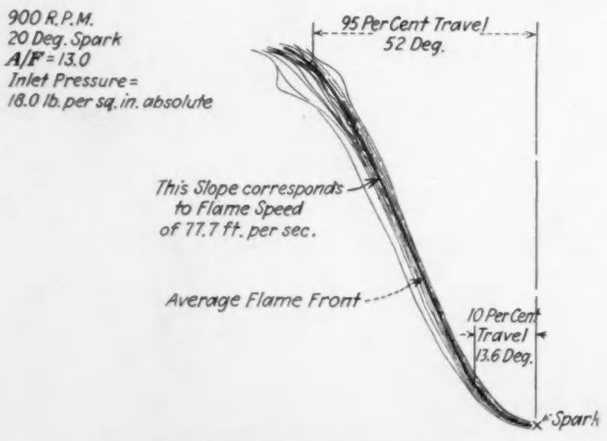
900 R.P.M.
20 Deg. Spark
A/F = 13.0
Inlet Pressure = 18.0 lb. per sq. in. absolute
95 Per Cent Travel 52 Deg.
This Slope corresponds to Flame Speed of 77.7 ft. per sec.
Average Flame Front
10 Per Cent Travel 13.6 Deg.
Spark

Fig. 5 – Superimposed Flame Fronts

The curve of flame speed against time, Fig. 4, can be divided roughly into three parts: an initial period of rapidly increasing velocity, a period of more or less constant flame speed and, near the end of the flame travel, a period of decreasing velocity. It is evident that no single measurement of time or velocity will serve to represent such a curve. After due consideration, however, it was decided that, for all practical purposes, measurements of the time occupied by the initial period, the total time for the flame to travel across the combustion-chamber, and the maximum speed of the flame in the middle period, would be satisfactory.

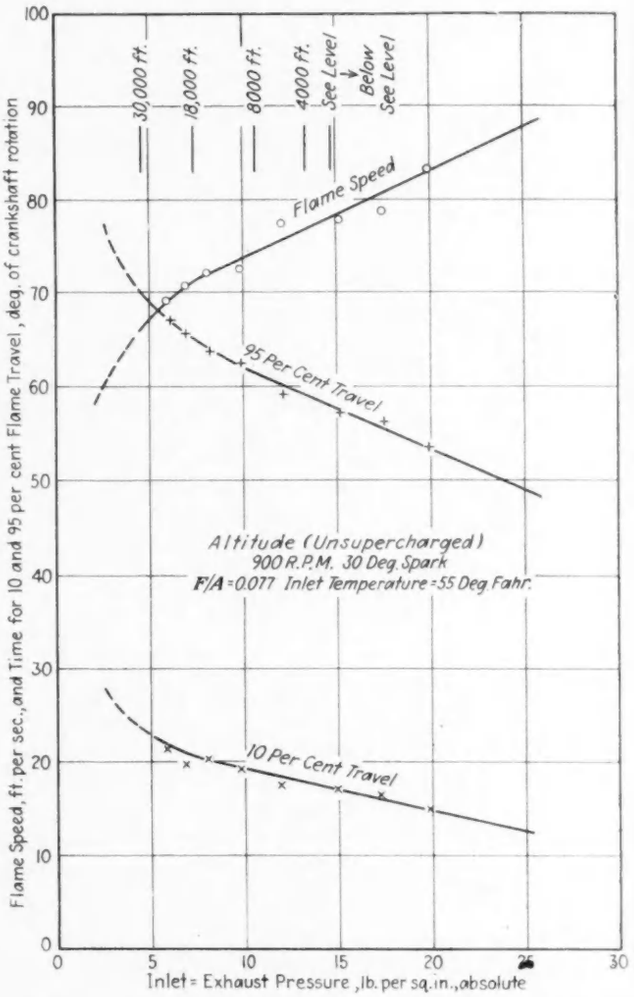
30,000 ft.
18,000 ft.
8000 ft.
4000 ft.
See Level
Below See Level
Flame Speed
95 Per Cent Travel
Altitude (Unsupercharged)
900 R.P.M. 30 Deg. Spark
F/A = 0.077 Inlet Temperature = 55 Deg Fahr.
10 Per Cent Travel
Flame Speed, ft. per sec., and Time for 10 and 95 per cent Flame Travel, deg. of crankshaft rotation
Inlet = Exhaust Pressure, lb. per sq. in., absolute

Fig. 6 – Variation of Flame Speed, 10 Per Cent Time, and 95 Per Cent Time with Altitude

From a study of a large number of the flame photographs, it was found that, under practically all conditions, the initial period of slow burning at the start of combustion occupied about 10 per cent of the distance across the combustion-chamber. The time occupied for the initial 10 per cent of flame travel was, therefore, used as a measure of this phase. For the main part of the curve, the average flame speed, as determined by the slope of the curve, was chosen. Since flame speed decreases rapidly in approaching the opposite side of the combustion-chamber, it is difficult to pick out the point on the film where the flame has just reached the cylinder wall. It was, therefore, necessary to choose some arbitrary distance nearly equal to the total distance across the combustion space to measure the time required for completion of the process. 95 per cent of the total length of flame travel was chosen finally for this purpose (line *C-D*, Fig. 4).

In order to reduce errors arising from the variation between cycles, about 15 flame photographs were used to determine the time and speed values for each run. The flame fronts of each group of films were superimposed by tracing, and the average flame front, as judged by eye, was taken as representative of the run. Fig. 5 shows a set of superimposed flame fronts.

Throughout this work an attempt was made to isolate each variable, and to find its effect alone on flame propagation. The values of the usual engine operating conditions were as follows: 900 r.p.m., full throttle, 30-deg. spark advance, 0.077 fuel-air ratio, 55 deg. fahr. inlet-air temperature, and 180 deg. fahr. water-jacket temperature. When any of these values was varied to determine its effect on flame travel, the others were held constant at the magnitudes stated above. The exhaust and inlet pressures used for the altitude runs correspond to those of the N.A.C.A. standard atmosphere. Supercharging, and the low pressures required for exhaust at various altitudes, were provided by a separately driven blower and exhauster.

Results

Altitude runs, unsupercharged. – Fig. 6 shows the variation of flame speed, 10 per cent time and 95 per cent time with varying altitude.

Since inlet and exhaust pressures are equal, the percentage of residual gases contaminating the fresh charge remains almost constant for any altitude. Consequently, the curves of Fig. 6 represent the effect of pressure alone on flame movement. As altitude increases, the flame speed decreases, whereas the 10 per cent and 95 per cent time increase. The bomb experiments by Stevens[6] and by Woodbury, Lewis and Canby[10] failed to show any effect of pressure on flame movement. Fiock[7] noted a pressure effect at very low pressures in CO-O_2 mixtures, flame speed increasing with pressure, as in the present tests.

Altitude Runs, Supercharged

The curves of Fig. 7 show the variation in flame speed and combustion time when a constant inlet pressure is maintained and exhaust occurs against a decreasing back pressure. The increase in flame speed and reduction in combustion time which occur as exhaust pressure is reduced are undoubtedly due to the decreasing proportion of residual exhaust gases in the mixture. This result agrees with the work of Stevens[3] and Fiock[9] who have shown that the effect of decreasing the concentration of inert gases is to increase the flame speed.

Throttling and supercharging at altitude. – The effect of varying the intake pressure at a given altitude from a closed-throttle condition to considerable supercharge is shown by Fig. 8. With constant exhaust pressure, flame speed increases, and time to burn decreases, as inlet pressure rises. These effects are evidently due to the combined influence of varying pressure (Fig. 6) and varying proportion of residual gases (Fig. 7).

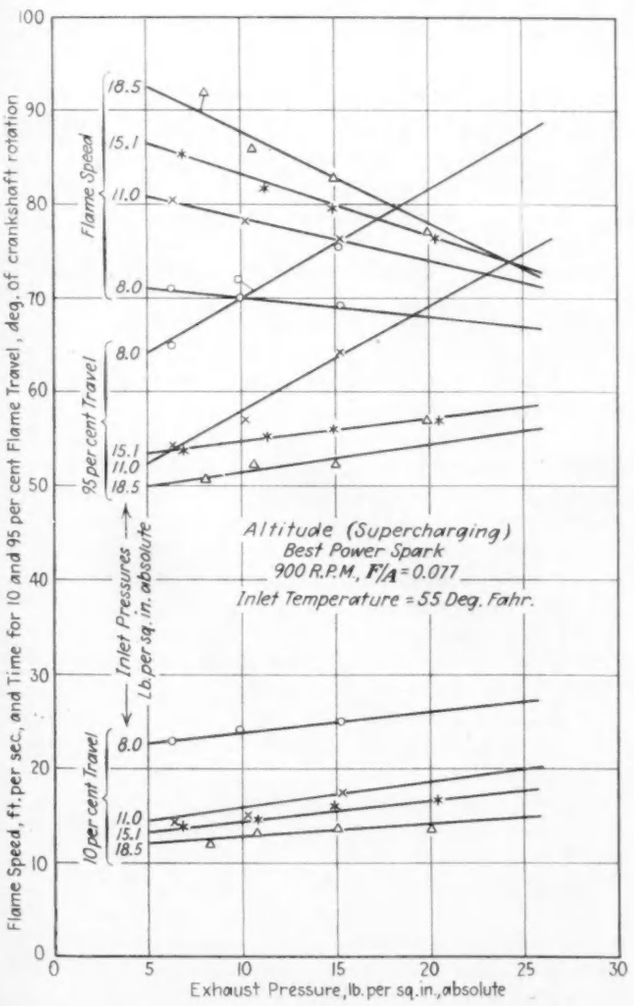
Flame Speed, ft. per sec., and Time for 10 and 95 per cent Flame Travel, deg. of crankshaft rotation
Exhaust Pressure, lb. per sq. in., absolute
Flame Speed
95 per cent Travel
10 per cent Travel
Inlet Pressures → Lb. per sq. in. absolute
18.5
15.1
11.0
8.0
Altitude (Supercharging)
Best Power Spark
900 R.P.M., F/A = 0.077
Inlet Temperature = 55 Deg. Fahr.

Fig. 7 – Variation of Flame Speed and Combustion Time with Exhaust Pressure

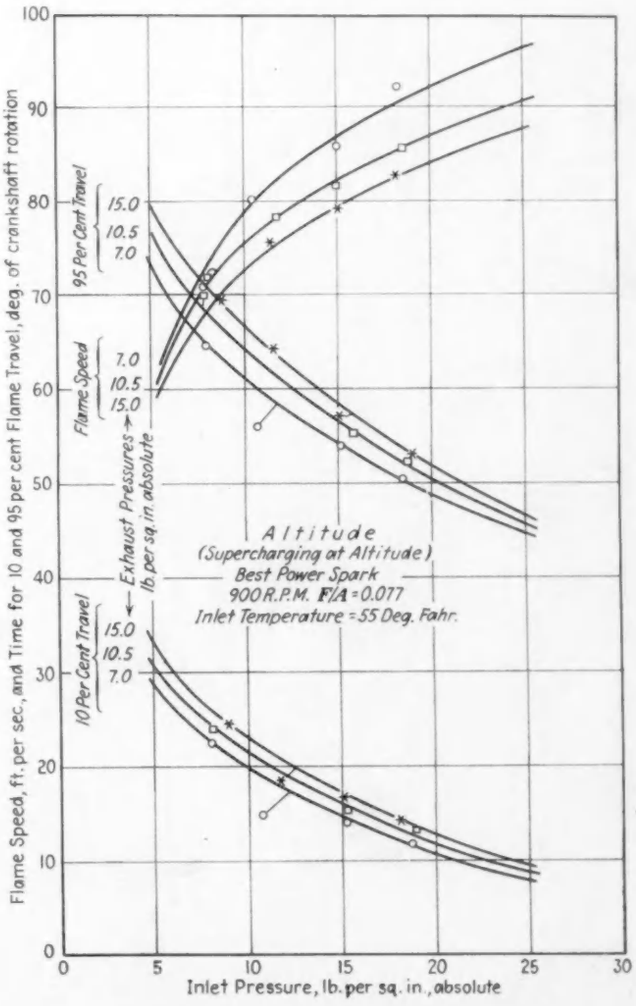
Flame Speed, ft. per sec., and Time for 10 and 95 per cent Flame Travel, deg. of crankshaft rotation
Inlet Pressure, lb. per sq. in., absolute
95 Per Cent Travel
Flame Speed
10 Per Cent Travel
Exhaust Pressures → lb. per sq. in. absolute
15.0
10.5
7.0
Altitude
(Supercharging at Altitude)
Best Power Spark
900 R.P.M. F/A = 0.077
Inlet Temperature = 55 Deg. Fahr.

Fig. 8 – Variation of Flame Speed and Combustion Time with Inlet Pressure

Inlet Temperature.–[19] Fig. 9 shows that the flame speed decreases, while the delay and combustion time increase, with increasing inlet-air temperature. This result was somewhat unexpected in view of the fact that the rate of most chemical reactions increases with increasing temperature. In the present instance, the decrease of flame velocity with increasing initial temperature may be due to the increased viscosity of the gases acting to damp the small-scale turbulence before ignition takes place. Of previous investigators, Duchene[12] alone seems to have observed flame speed with varying initial temperature, and he found no appreciable effect.

Humidity

Although it has been shown by Fiock and King[7] that flame speed increases with increasing concentration of water vapor in CO-O_2 mixtures, the results obtained in this investigation show that the effect of increasing the humidity of the air is to decrease flame speed (Fig. 10). This effect is confirmed by general experience in operating engines, which shows that increasing spark-advance is necessary with increasing humidity.[20] Probably the disagreement with Fiock's results is attributable to the difference in fuel composition.

Engine speed.–In order to isolate engine speed as a variable, it was necessary to hold volumetric efficiency constant by varying the inlet pressure so as to maintain a constant air consumption per revolution. Fig. 11 indicates that flame speed varies almost linearly with engine speed. This result is in agreement with previous investigations.[14, 16, 18] The curve of flame speed, if projected, would not pass through the origin at zero engine speed because the stagnant mixture of fuel and air has a definite flame speed.

Although the 10 per cent period expressed in seconds is shorter as engine speed increases, we see from Fig. 11 that, if this period is plotted in degrees of crankshaft rotation, it increases with engine speed.

In a similar manner, the time for the flame to travel 95 per cent of its total distance, measured in degrees of crankshaft rotation, increases with engine speed. Apparently the necessity of increasing spark advance with increasing engine speed is largely due to the period of initial combustion, as suggested by Ricardo.[21]

The rapid increase in the maximum rate of combustion with engine speed appears to be due primarily to an increase in small-scale turbulence resulting from higher gas velocities through the intake system. The marked effect of turbulence on flame speed in constant-volume bombs has been noted by many investigators.

Fuel-air ratio.–The effect of changing the mixture ratio on flame travel in an engine may be seen from the curves in Fig. 12. The flame speed is maximum at a mixture corresponding approximately to the best-power mixture. Time for 10 per cent and 95 per cent flame travel both have their minimum value at approximately the best-power mixtures. These results are similar to those obtained in many previous investigations, both in bombs and in engines.

[19] The two temperature runs were made using an intake manifold with less restrictions than that used for the other runs. This arrangement resulted in an increase in the air capacity and, hence, in the cylinder pressure. Because of the increased pressure, the flame speed with atmospheric inlet conditions is higher in the temperature runs than in any of the other runs.

[20] See N.A.C.A. Technical Note No. 309, June, 1929: "Correcting Engine Tests for Humidity," by D. B. Brooks.

[21] See "Engines of High Output," by H. R. Ricardo, Van Nostrand, 1927.

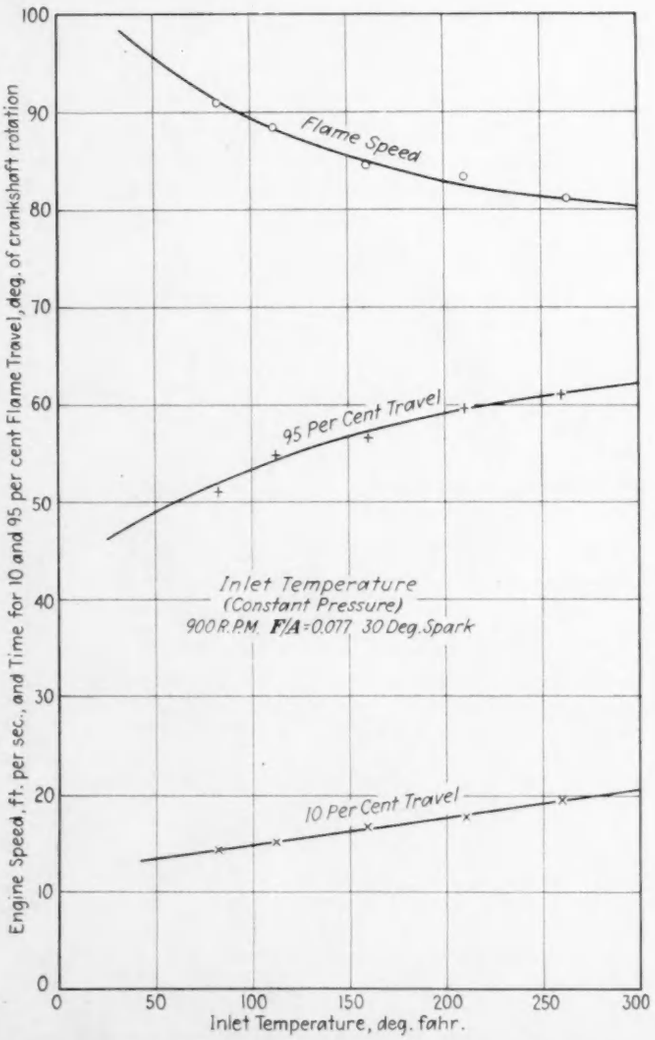

Fig. 9 – Variation of Flame Speed and Combustion Time with Inlet Temperature

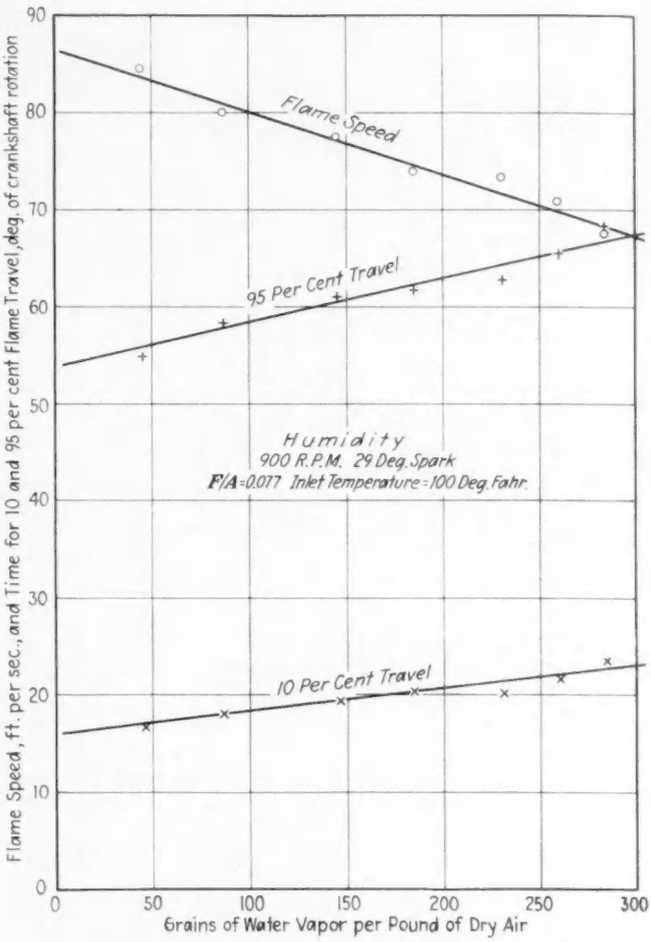

Fig. 10 – Variation of Flame Speed and Combustion Time with Humidity

Ignition timing. – The curves of flame movement vs. ignition timing (Fig. 13) show that spark timing has considerable effect on maximum flame speed. This is probably a pressure effect, the maximum flame speed occurring when the piston reaches top-center about in the middle of the combustion process.

The length of the 10 per cent period shows a steady, although slight, increase with advancing spark. This result may be explained by the fact that, as the spark is advanced, the burning during this period occurs at lower pressure.

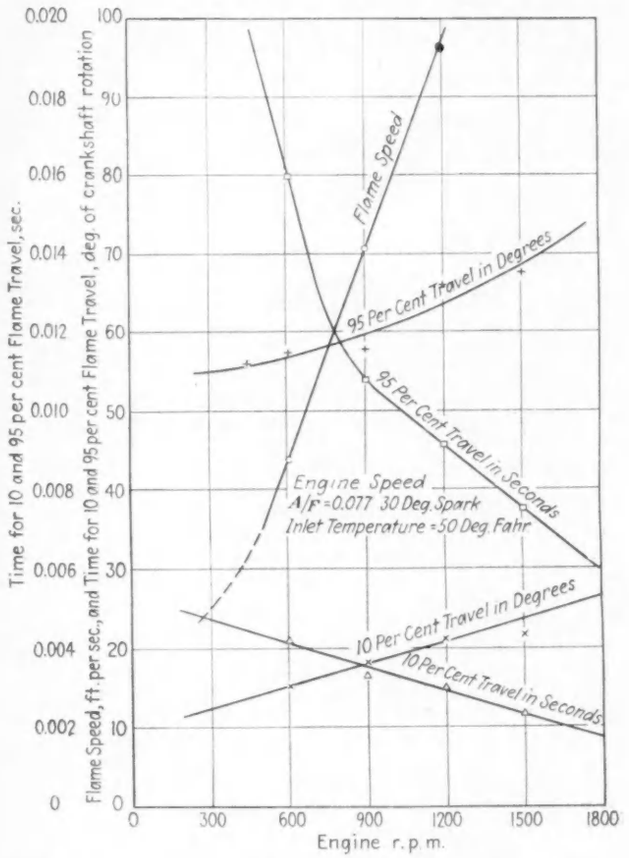
Time for 10 and 95 per cent Flame Travel, sec.
Flame Speed, ft. per sec., and Time for 10 and 95 per cent Flame Travel, deg. of crankshaft rotation
Engine r.p.m.
Flame Speed
95 Per Cent Travel in Degrees
95 Per Cent Travel in Seconds
Engine Speed
A/F = 0.077 30 Deg. Spark
Inlet Temperature = 50 Deg. Fahr.
10 Per Cent Travel in Degrees
10 Per Cent Travel in Seconds

Fig. 11 – Variation of Flame Speed and Combustion Time with Engine Speed

Conclusions

(1) Under conditions of normal combustion without detonation, the general nature of the movement of the flame front remains the same, that is, a period of slow burning exists after the initiation of combustion during which the flame speed is gradually increasing until it reaches a maximum at a point approximately 10 per cent of the distance across the combustion-chamber. The flame front then continues to travel on at a nearly uniform maximum speed to a point approximately 90 per cent of the distance across the chamber. From this point, as the flame approaches the opposite side of the chamber, its speed decreases rapidly.

(2) The speed of the flame front increases, and the time for 10 and 95 per cent flame travel are reduced by the following:

Increase in the pressure level at which combustion occurs.
Decrease in initial temperature.
Decrease in proportion of residual gas.
Increase in small-scale turbulence.

(3) The average flame speeds during the initial period, the period of rapid flame travel, and the period of low flame speed near the end of the process, tend to vary in the same direction over a wide range of engine variables.

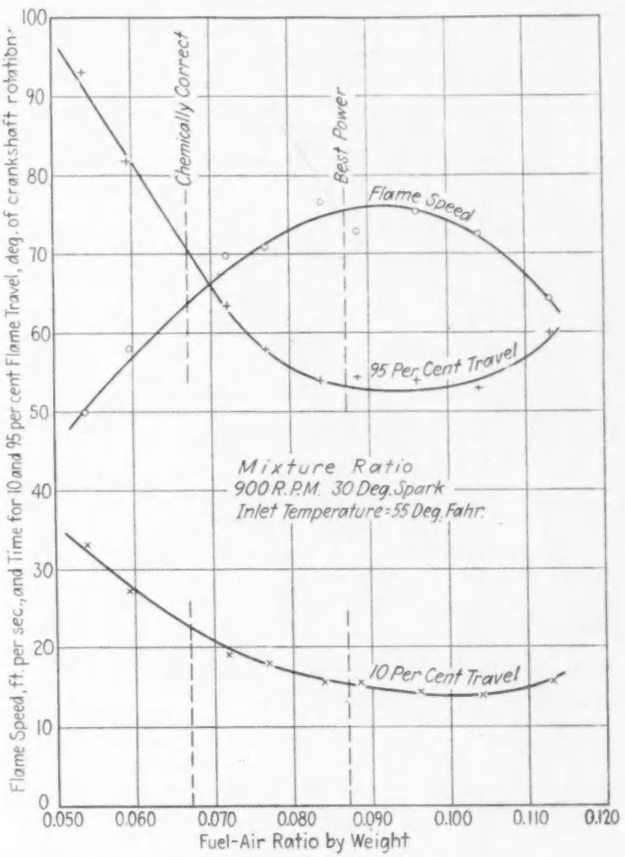
Flame Speed, ft. per sec., and Time for 10 and 95 per cent Flame Travel, deg. of crankshaft rotation
Fuel-Air Ratio by Weight
Chemically Correct
Best Power
Flame Speed
95 Per Cent Travel
Mixture Ratio
900 R.P.M. 30 Deg. Spark
Inlet Temperature = 55 Deg. Fahr.
10 Per Cent Travel

Fig. 12 – Variation of Flame Speed and Combustion Time with Mixture Ratio

(4) Under most operating conditions, the initial period of slow flame travel occupies from 25 to 30 per cent of the total time required for combustion.

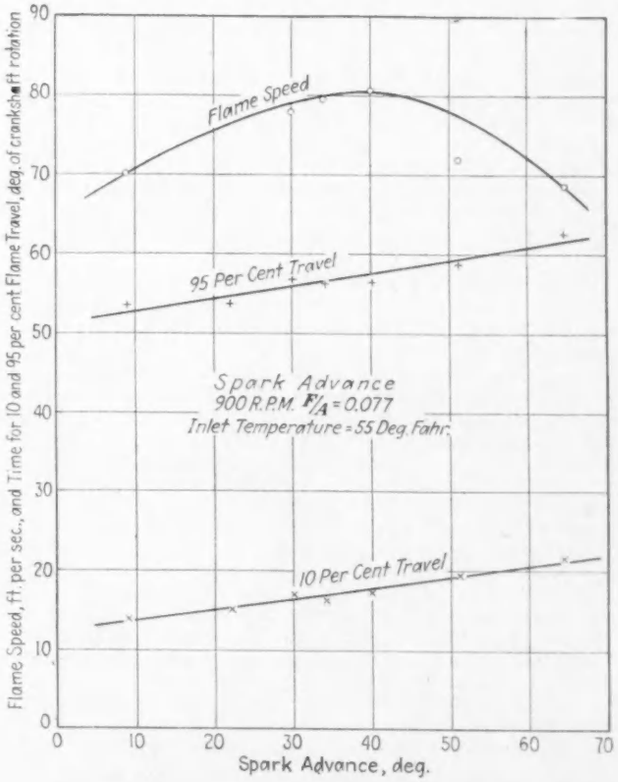
Flame Speed, ft. per sec., and Time for 10 and 95 per cent Flame Travel, deg. of crankshaft rotation
Spark Advance, deg.
Flame Speed
95 Per Cent Travel
Spark Advance
900 R.P.M. F/A = 0.077
Inlet Temperature = 55 Deg. Fahr.
10 Per Cent Travel

Fig. 13 – Variation of Flame Speed and Combustion Time with Ignition Timing

Discussion

Effect of Combustion-Chamber on Flame Progress

– L. C. Lichty
Yale University

THE paper by Bouchard, Taylor, and Taylor is most interesting particularly from the standpoint of the effect of pressure and temperature upon flame progress. Although the experiments deal with one type of combustion-chamber, it may be of interest to point out some of the theoretical as well as practical effects of different types of combustion-chambers upon flame progress.

The rate at which the flame travels from the spark-plug to the furthermost point in the combustion-chamber of the internal-combustion engine for any given set of conditions depends upon two factors:

(1) The rate at which the flame progresses into the unburned relative to the unburned.

(2) The rate at which it is propelled due to the expansion of the burned portion.

The first rate is the velocity of the flame one would observe if moving with the same velocity as a thin layer of unburned through which the flame is progressing. Stevens[a] found this rate to be linear and constant for a given mixture and independent of pressure in his constant-pressure experiments. He also stated[a] that this rate was "constant during the reaction" for the constant-volume combustion process.

The second rate depends upon the relative volume of burned gases back of the flame front. The first portion of the charge to burn occupies initially about five times the volume of the equivalent last portion of the charge when the flame reaches it. Since both portions rise in temperature about the same amount during the reaction, the relative increase in volume is about the same. This condition makes the expanded volume of the first portion much greater than the last. Thus, at the beginning of the combustion process, most of the flame movement is due to the expansion of the burned, and this rate approaches zero toward the end of the flame travel, at which time the flame progresses with the relative rate just mentioned.

The effect of combustion-chamber design on rate of flame travel can be illustrated partially in comparing several combustion-chambers, two of which are not applicable to engines. In a long-tube combustion-chamber the volume of the burned back of the flame front when it has traveled one-fourth the length of the tube is nearly 25 per cent. In the combustion chamber of the authors (Bouchard, Taylor, and Taylor) there is about 17 per cent burned volume back of the flame front for the same relative flame travel. The flat cylindrical valve-in-head combustion-chamber with the spark-plug on one side has about 11 per cent burned volume for the same relative flame front position. In the spherical bomb fired at the center there will be only 1.56 per cent volume of burned when the flame has traveled one-quarter of the way to the walls of the bomb.

Thus, the relative time required for the flame to travel the first fourth of the distance would be least in the spherical bomb and most in the tube with the other two combustion-chambers in between, as indicated by the volume back of the flame front.

There are two other factors which affect the flame progress in the engine cylinder for a given set of conditions, namely, heat transfer and movement of the piston. Heat transfer from the charge in its early stages of burning must be fairly high on a percentage basis since the mass of the burned is small compared to the surface of the flame and the walls the hot gases contact. Thus, any appreciable heat loss from the burned portion will retard the movement of the flame.

In the authors' engine the movement of the piston near dead-center would force some of the charge into the valve pocket, which action also would retard the movement of the flame. This effect can be seen by imagining no flame progress due to burning while the piston moves up in the stroke near dead-center and with the flame in the valve pocket. This condition would result in the flame front being pushed back toward the spark-plug. Hence, piston motion upward with flame progress will retard the motion of the flame in this type of combustion-chamber. Piston motion in the valve-in-head type of cylinder would have little effect on displacing the flame position.

Thus it appears that combustion-chamber shape, spark-plug location, heat transfer, and piston motion are also factors which influence flame travel in the engine cylinder.

The experiments dealing with the variation in inlet temperature show a decrease in flame speed with an increase in inlet temperature. This is to be expected from the standpoint that an increase in inlet temperature reduces the quantity of the fresh charge and increases the percentage dilution with the clearance gases. Dilution should decrease the flame speed. If increasing inlet temperature has any effect in increasing the flame speed in the experiments reported, it is apparently masked completely by the effect of the dilution. This is one of the inherent difficulties in running experiments on flame progress in the internal-combustion engine.

The fact that an increase in temperature increases the speed of reaction in chemical processes may have little bearing on the rate at which a flame is propagated in a combustion-chamber so long as the temperature of the unburned mixture ahead of the flame front does not reach the temperature at which flame will develop spontaneously.

[a] See N.A.C.A. Technical Report No. 372, 1931: "The Gaseous Explosive Reaction – The Effect of Pressure on the Rate of Propagation at the Reaction Zone," By F. W. Stevens.

Further Explanation of Authors' Results

– Charles F. Marvin, Jr.
Mechanical Engineer, National Bureau of Standards

I HAVE read this paper with great interest and was gratified to find the many points of agreement with results obtained at the National Bureau of Standards. The data regarding the effects of pressure (Fig. 6) and of temperature (Fig. 9) on flame speed are particularly valuable for, despite the importance of these variables, little or no reliable experimental information concerning them has heretofore been available. (Figure numbers refer to those in Messrs. Bouchard, Taylor, and Taylor's paper.)

Increased pressure of the charge during the combustion period is certainly the only obvious cause of the increase in flame speed shown in Fig. 6. However, it is conceivable that small changes in charge temperature, dilution, and turbulence may have combined to influence the results.

In seeking an explanation for the decrease in flame speed shown in Fig. 9, it must be remembered that, with inlet and exhaust pressure constant, an increase in inlet air temperature causes a decrease in the weight of charge drawn in per cycle. Since the pressure of the residual gas remains constant and its temperature probably decreases, there will be a slightly larger weight of residual in the clearance space to mix with this smaller weight of fresh charge. Thus, as inlet air temperature is increased, there will be an increase in the proportion of exhaust gases in the explosive charge at ignition. Due to its greater dilution and higher temperature, the warmer charge will expand less when burned, and compression of the unburned gases by the advancing flame will not proceed as fast or as far as with a cooler charge. Thus, the rise in inlet temperature of Fig. 9 causes an increase in the dilution and a decrease in the average pressure of the charge during the combustion period, which apparently more than offsets any increase in flame speed which may result from increased charge temperature. If special precautions had not been taken to maintain the fuel-air ratio at 0.077, an enrichment of the mixture would have added to the uncertainties of interpretation.

It is very difficult, if not impossible, to isolate completely the independent effects of pressure and temperature on flame speed in an engine. Moreover, measurements of the spread of flame in an engine cannot be analyzed to yield accurate values of the speed of propagation into the unburned charge, which is the fundamental speed of greatest interest. However, this fundamental "transformation velocity" can be calculated from observations of the spread of flame and the development of pressure in a spherical bomb with central ignition. The National Bureau of Standards, with the support of the National Advisory Committee for Aeronautics, is now constructing apparatus for this purpose, and it is hoped that experimental data showing the separate effects of temperature and pressure on flame speeds will be obtained during the coming year.

Erratum

A credit line was omitted from the bottom of The Viscosity Conversion Chart, published as Fig. 3 of the paper: "Crankcase-Oil Temperature Control," by Ellis W. Templin, on page 328 of the August issue of the SAE JOURNAL. With the exception of the formula on the lower-right-hand side, the chart is a copy of the one appearing on page 34 of the July 10, 1935, issue of *National Petroleum News*, and originated by M. G. Van Voorhis, an engineer of its staff.

A Spark-Advance Indicator and Knock-Rating Observations

By Sidney Oldberg, Gilbert Way, and J. B. Macauley
Chrysler Corp.

THE increased sensitivity of the modern high-compression engine to ignition timing, together with the use of vacuum-controlled distributor advance, has emphasized the need of an accurate spark-advance indicator for road-test use. In addition to the fact that spark advance occurring on the road cannot be predetermined in the laboratory, other causes of discrepancy between road and laboratory results may occur.

An apparatus using purely electrical means for indicating spark advance on the road is described in detail as to principles involved, design, application, and operation. Curves showing spark-advance data obtained in cars with the instrument as compared to laboratory data on the same distributors are included.

The use of such an instrument in obtaining octane requirements on the road is self-evident, especially when a sensitivity of up to 4 octane numbers per degree of spark advance has been observed in current-model cars. In an effort to increase the accuracy of fuel road ratings, a series of five fuels were road-rated on several cars of the same model. The data obtained have been tabulated, with trends based thereon indicated, and are included as a supplementary progress report.

THE timing of ignition in the operating cycle of an internal-combustion engine has long been recognized as one of the major factors affecting its power output, economy, and knocking characteristics. As fuels of uniformly high antiknock quality became distributed widely, the engine designer was able to narrow down considerably on his compromises involving compression ratio and spark advance, a fact which had a large influence in the development of today's high-compression engine. Greater sensitivity to variations in spark timing has accompanied this development, and optimum performance on fuels of a given octane number requires a closer specification of the ignition-timing range under operating conditions.

Manufacturers of ignition equipment have devised spark-control mechanisms which, under proper conditions, reproduce the timing requirements of the engine. The production tolerance of ± 2 deg., however, which is in general practice today, is not in line with the increased spark sensitivity of the modern engine. A recent survey has stated that the octane requirement of a 1936 model car may change as much as 4 octane numbers per degree of spark advance; thus a possible variation of up to 16 octane numbers in fuel requirement might be included in the range of low- to high-limit distributors, in addition to a noticeable effect upon car performance.

Laboratory spark-advance indicators, usually of the neon-tube variety, have been in general use for many years on engine test stands and other stationary units. The accurate indication of spark advance under road operating conditions has not, to our knowledge, been accomplished successfully in a practical manner.

With the use of vacuum control of distributor advance, the prediction of ignition timing occurring in road operation cannot be made from laboratory data, even under the most favorable conditions. The fluctuation of vacuum to which the vacuum-controlled advance is subjected depends not only upon variable operating conditions, but is also a function of a delicate relationship between the carburetor throttle blade and the location of the vacuum take-off port which, in turn, may be affected by slight deposits of dust or gum.

Among the other causes of discrepancy between laboratory tests and actual road spark advance may be:

(1) Sticking or lagging action of the governor mechanism.

(2) Sticking of the vacuum-advance mechanism.

(3) Lash and variation in gears, timing-chain whip, and concentricity of distributor driveshaft.

(4) Slippage or error in static setting of the distributor.

In designing an instrument for road use, it was recognized early that any form of accessory drive from the crank, cam, or distributor shafts introduced errors in the determination of top dead-center that eliminated such drives from serious consideration, so a purely electrical method was decided upon.

Fundamentally, the system adopted is as follows: At a definite point of the cycle – in this case, 60 deg. before top dead-center – a relay is tripped which instantly allows current to pass through a direct-current meter. The occurrence of the spark at some later point trips another relay which stops the passage of current, and the cycle is repeated *every* engine revolution. Fig. 1 is an oscillogram of the current cycle. If the meter is damped then, over the operating speed range of

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., May 7, 1937.]

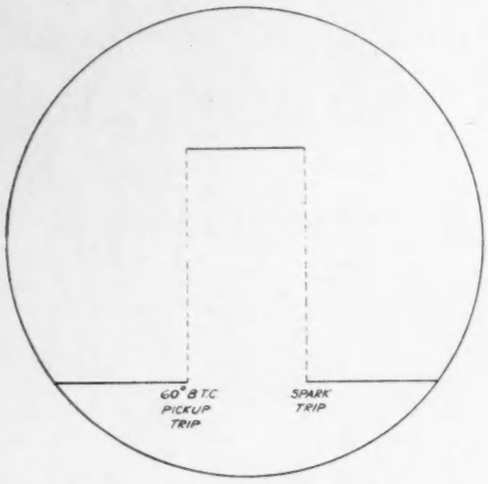

Fig. 1 – Oscillogram of Current Pulse Through Meter

the car, the average value of current as indicated by the meter is in the same proportion to the maximum current which flows between the events, as the interval between the events is to 360 deg. Thus the meter indicates the angle between these events.

This method is generally applicable to the measurement of angular duration where suitable pickups may be used to indicate the limits of the events being investigated.

The fixed tripping point (60 deg. before top dead-center) is actuated by an electromagnetic pickup working from a stud in the flywheel (Fig. 2). The stud rotates past the pole of a permanent magnet which carries a winding, thus closing a magnetic circuit which goes through the pole, bracket, motor, flywheel, and stud. (A gap of about 1/16 in. is used, although the operation is not critical to this gap.) A voltage is thus induced in the coil which goes slowly positive and very rapidly negative as the centerline of the stud passes that of the pole piece, then drops to zero as shown in Fig. 3.

Referring to the wiring diagram of Fig. 4, this voltage is amplified and applied to the grid of a gas triode (RCA 885). This tube has the property of passing no current from plate to cathode unless the grid bias is reduced below a certain value which depends on the plate voltage but, once current is started, the current is independent of the grid voltage. The current flow is stopped by raising the cathode potential to some value positive with respect to the plate for an instant, thus allowing the grid to regain control. Thus, in the present

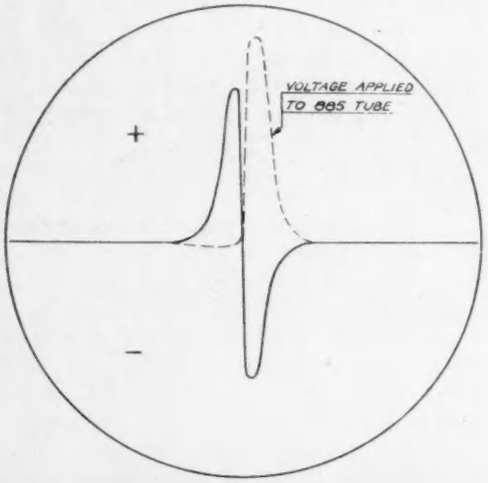

Fig. 3 – Oscillogram Showing Wave Form of Voltage Output from Magnetic Trip Coil

apparatus, the amplified negative side of the pulse generated by the flywheel pickup reduces the bias on the 885 and permits it to conduct current through the meter, a 0 – 1 milliammeter.

In the spark trip circuit, the primary of the ignition system is used to eliminate electrostatic pickup in the internal circuits. The oscillation occurring with the spark is rectified and applied to the second 885, thus firing it simultaneously with the spark. (Only the first spark occurring after the firing of the first 885 takes part in the process, the advance being thus the average of No. 1 and No. 6 cylinders on a six-cylinder engine, or No. 1 and No. 8 cylinders on an eight-cylinder engine.) This action raises the cathode potential of the first 885 higher than its plate, thus cutting off the current flow through the meter circuit.

Calibration of the instrument is made directly by measuring the current in the meter circuit with the spark trip not operating. This is the condition when the selector switch is in the No. 2 position. The current is now that which would flow if the interval between the pickup pulse and the spark were 360 deg., and this value is adjusted to full scale on the

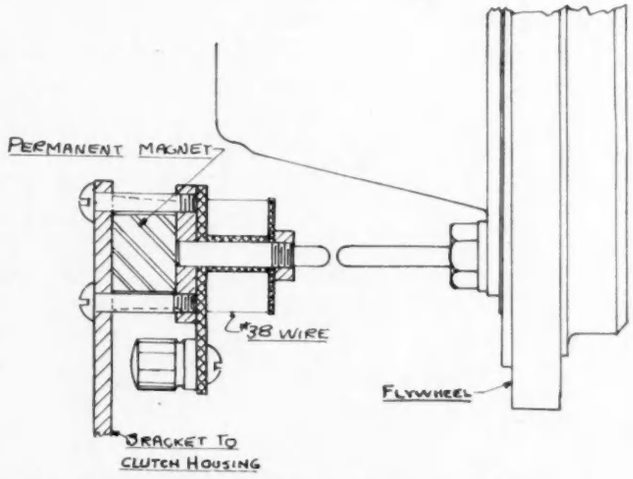

Fig. 2 – Flywheel Pickup for Spark-Advance Indicator

meter, (1 mil. amp.) by the rheostat, the meter being shunted by a resistance equal to $\frac{7}{29}\left(\frac{70}{360-70}\right)$ that of the meter for a 70-deg. operating scale.

The third position of the selector switch puts both trips in operation but retains the meter shunt. This arrangement shows the operator whether the circuits are working properly, under which condition the needle reads between zero and approximately a sixth of full scale. This is a precautionary measure as failure of the spark trip to stop the meter circuit without a shunt would overload the meter some 500 per cent. The fourth position removes the shunt and the meter reads the spark advance.

Installation on a test car requires only the removal of the clutch dust pan, replacing of a flywheel capscrew with the stud, installation of a bracket holding the pickup, and taking leads from the instrument to 6 volts, ground, ignition primary, and flywheel pickup.

Synchronization of the magnetic pickup unit to trip at 60 deg. before top dead-center is accomplished by properly locating this unit by means of its supporting bracket. This process is simplified by setting the engine on top dead-center and then mounting the pickup directly in line with the flywheel stud. The stud can then be moved one hole ahead which, in

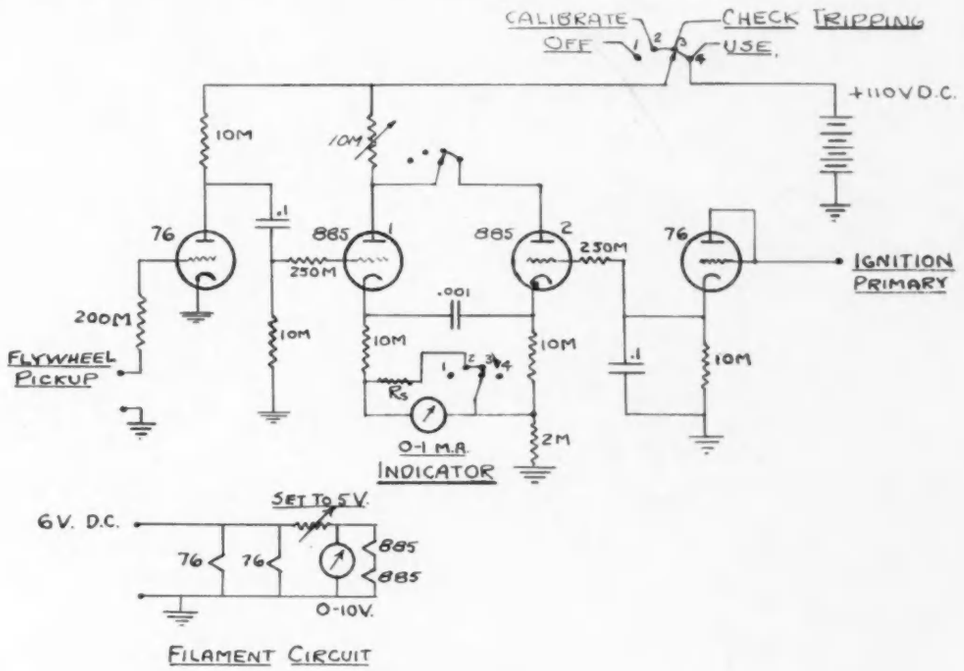

Fig. 4 – Direct-Reading Spark-Advance Indicator

the case of the ordinary six-hole clutch assembly, will sweep by the pickup at 60 deg. before top dead-center.

Operating Characteristics

After making the various connections, the switch is turned to the second point and the meter deflection adjusted to full scale. This adjustment determines the maximum value of the current pulse and must be set accurately as any error affects the reading in direct proportion to the meter deflection. This adjustment also compensates automatically for variations in tube characteristics and battery voltage. The next point on the switch shows the operation without subjecting the meter to overload. If the meter needle remains at zero the magnetic trip is not operating. If the needle goes to full scale, the spark trip is not operating and, if the needle reads below one-sixth full scale but occasionally kicks up, it indicates an intermittent failure of the spark trip, which may be due to an arc being drawn across the distributor points. Finally, if the meter reads steadily at less than the one-sixth full scale, everything is in order, and the shunt may be removed from the meter by switching to Position No. 4, and the spark advance will be indicated continuously on the instrument panel shown in Fig. 5.

At present, there are a few limitations as to the accuracy of the apparatus. It is believed that, with additional development, the slight deviations from theoretical operation which now necessitate corrections can be minimized to negligible values.

The accuracy of the instrument is limited, first, to the accuracy of the first tripping point (60 deg. before top dead-center). This is an electromagnetic unit and has some lag. Tests with a stroboscope show a lag in the present pickup unit of 1 deg., independent of speed.

A theoretical source of error is the time required for the current pulse to reach its maximum value, and to return to zero at the end of the pulse. This error is, for all practical purposes, negligible since the circuit is resistive and the breakdown of the relay tubes is instantaneous. The maximum possible variation due to this cause would be about 0.04 deg.

The meter itself may introduce a small error. The meter in the present instrument is rated within 1 per cent of accuracy.

At low speeds the meter attempts to follow the wave form of the pulse causing the needle to oscillate over several degrees. At present it is necessary for the operator to average this movement to obtain the reading, but this fluctuation can be minimized by the use of a more highly damped meter.

The accuracy of the apparatus was checked first on laboratory engines equipped with neon spark indicators. It was noticed that the spark advance read on the meter was slightly higher than that shown by the neon tube, indicating that either the magnetic trip was lagging or the spark trip was occurring too soon. Stroboscopic investigation showed the

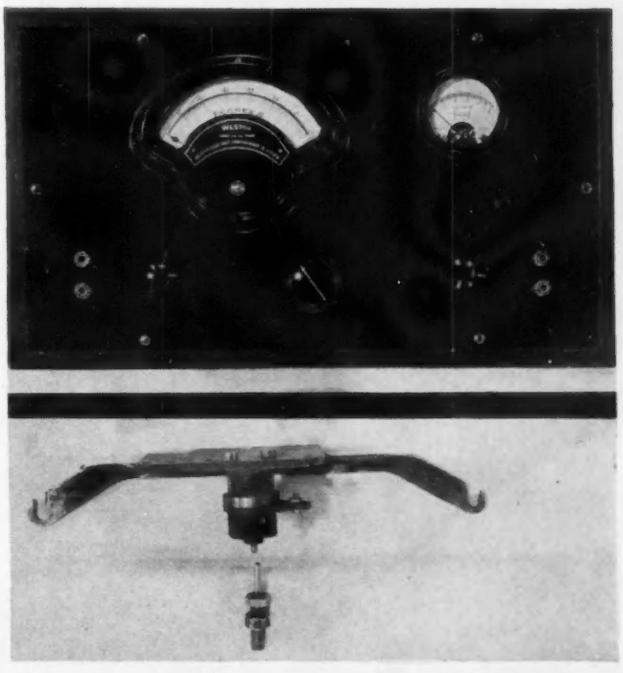

Fig. 5 – Panel and Pickup of Spark-Advance Indicator

Table 1 – Car Road Ratings in 1937 Six-Cylinder Cars of Same Make and Model

	CAR No. 1 Barometer 29.10 Air temp. 62°, 65½°* Rel. hum. 55%, 41%*				CAR No. 2 Barometer 29.15 Air temp. 63° Rel. hum 64%				CAR No. 3 Barometer 29.44 Air temp. 66° Rel. hum. 34%				CAR No. 4 Barometer 29.50 Air temp. 63½° Rel. hum. 22%			
	Dist. set.	Maximum knock intensity Speed	Miles carbon	Road rating	Dist. set.	Maximum knock intensity Speed	Miles carbon	Road rating	Dist. set.	Maximum knock intensity Speed	Miles carbon	Road rating	Dist. set.	Maximum knock intensity Speed	Miles carbon	Road rating
FUEL A																
Pa straight run	Std.	VL @ 23	2640	68½	2°Adv	VL @ 23	8700	68½	Std.	L @ 23	3830	68	6°Adv	L @ 25	2910	68½
2 cc. Pb per gal.	4°Adv	M @ 25*	2670*	68*	6°Adv	MH @ 23	8700	69½	4°Adv	M @ 23	3875	68	10°Adv	M @ 25	2960	68½
Ratings:									8°Adv	H @ 23	3925	67½				
M.M. 69 Res. 68½																
Road, average 68½																
FUEL B																
Mid. cont. str. run	Std.	T @ 20	2645	71					Std.	T @ 22	3835	71	6°Adv	VL @ 23	2915	72½
25% benzol	4°Adv	L @ 22	2675	72					4°Adv	VL @ 18	3880	72½	10°Adv	L @ 25	2970	75
Ratings:									8°Adv	MH @ 22	3900	73				
M.M. 71 Res. 74½																
Road, average 72½																
FUEL C																
Cal. CRK and str. run	Std.	T @ 20	2655	72					Std	VL @ 15	3840	69½	6°Adv	VL @ 20	2935	71
Ratings:	4°Adv	L @ 23	2680	71					4°Adv	L @ 20	3860	71	10°Adv	LM @ 25	2965	71
M.M. 69½ Res. 71									8°Adv	H @ 17	3900	69½				
Road, average 70½																
FUEL D																
100% mid. cont. vapor	Std.	No kn.							4°Adv	T @ 15	3895	76½	6°Adv	No kn.		
phase CRK	4°Adv	T @ 20	2685	75					8°Adv	MH @ 60	3925	73	10°Adv	T @ 23	2990	78½
Ratings:																
M.M. 70 Res. 78½																
Road, average 76																
FUEL E																
Commercial Q	Std.	VL @ 25	2670	68½	2°Adv	L @ 20	8700	67½	Std.	L @ 25	3855	68½	6°Adv	L @ 25	2890	68½
Ratings:	4°Adv	LM @ 25	2690	68	6°Adv	H @ 22	8700	68½	4°Adv	M @ 25	3870	68½	10°Adv	M @ 25	3000	69
M.M. 68½ Res. 70½									8°Adv	XH @ 20	3930	66½				
Road, average 68																

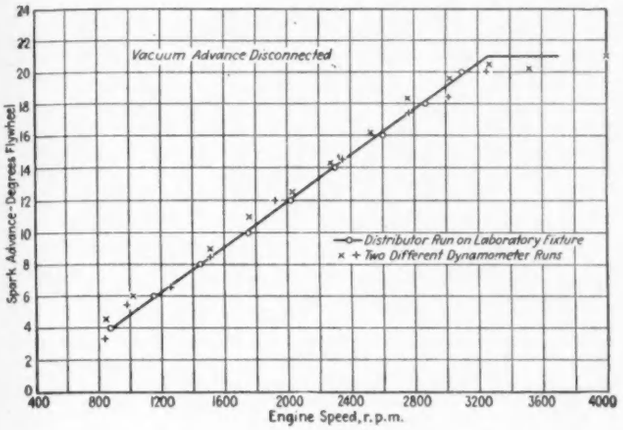
Vacuum Advance Disconnected
Distributor Run on Laboratory Fixture
Two Different Dynamometer Runs
Spark Advance - Degrees Flywheel
Engine Speed, r.p.m.

Fig. 6 – Road Spark-Advance Indicator – Recorded from Car No. 5 on Chassis Dynamometer

magnetic lag which, for the time being, was evaluated and a correction applied to subsequent meter readings. The corrected values now obtained coincided with the neon indicator throughout the advance and speed range within the accuracy of reading, about ¼ deg.

Fig. 6 shows the spark-advance points as read from the instrument installed in car No. 5 compared to the advance curve obtained on a laboratory fixture for determining distributor characteristics. Close agreement between the points obtained and the laboratory curve is evident and, from the results of check runs made on the dynamometer, we believe that the slight variations shown are not instrument or reading errors but are actual deviations occurring at the time that the readings were made.

The apparatus was installed later in car No. 2, and a complete record made of the spark indications at road loads, wide-open throttle acceleration, and constant speeds, with the distributor vacuum advance disconnected. The distributor was laboratory-checked for advance before and after the test, as well as the static setting in the engine. Fig. 7 shows the result obtained, the shaded areas being bounded by the values obtained from the laboratory fixture on subsequent checks. The full vacuum-advance position was plotted its full value of 21 deg. above the governor curve. The spark advance occurring during road-load operation shows considerable variation from one direction to the opposite because of a strong wind blowing during the test. Against the wind at 20 m.p.h. there was 10 deg. more spark advance than with it, due to the fact that the greater throttle opening in the former case had more nearly uncoverel the vacuum port in the carburetor, this port being so located to eliminate the vacuum advance under idling conditions. At high speeds the vacuum dropped off much more rapidly due to greater throttle opening being required against the wind. It is evident that these road-load spark-advance curves can be obtained only under the actual operating conditions on the road. The governor advance curves shown on the lower part of Fig. 7 indicate a variation in spark advance under accelerating and constant-speed conditions. This variation is due, in part, to a lag in the governor advance mechanism during acceleration and, to some extent, to angular variation between the crankshaft and distributor shaft under accelerating and constant-speed conditions. This spread in spark advance may be the reason for the higher knock intensities observed on the road at full-throttle constant-speed or decelerating conditions as compared to full-throttle acceleration. The variation between the road-advance and laboratory-advance curves through the higher speed range indicates either erratic governor characteristics or the occurrence of angular lag in the distributor shaft.

The desirability of such an instrument for use in determining octane requirements is self-evident. In an effort to increase the accuracy of fuel road ratings, a series of ratings were made on a group of fuels at various spark advances and knock intensities as follows:

Cars of the same make and model were used in an effort to determine the variation in knock ratings that might be expected in individual cars of the same model. Laboratory checks were made on each distributor and are shown in Figs. 8 and 9, with the relation of engine speed to car speed shown in Fig. 10. It was originally planned to use the road spark indicator in each of the cars to check the road spark advance. This checking was done in cars No. 1 and 2 (Fig. 7), but we were unable to complete this program due to lack of time.

Table 1 gives the results of these road ratings. Where car sensitivity to octane number would permit, ratings were made with reference fuel brackets of 5 per cent *C*, and two of the three observers rated the fuels blindly. Unfortunately, almost all of the cars available which were suitable for fuel ratings were comparatively new, with low mileages. Car No. 2 was too far in the opposite direction, being a 1937 model with 35,000 miles already on the engine, 8700 miles since last carbon removal, but only 100 miles since experimental piston-rings were installed. Because of poor engine condition, this car was regarded as unsuitable for reliable fuel ratings. Results on the two fuels rated, however, are shown in the table.

Assuming the accuracy of road ratings to be within 1 octane number, the following indications may be drawn from this data:

(1) Fuels of low sensitivity such as Fuels *A*, *C*, and *E*,[1] rate substantially the same regardless of spark advance, knock intensity, or individual car used. Only one exception was noted – Fuel *E* in car No. 3 dropped 2 octane numbers when rated at extra-heavy knock intensity, but ratings in this range are generally considered unreliable.

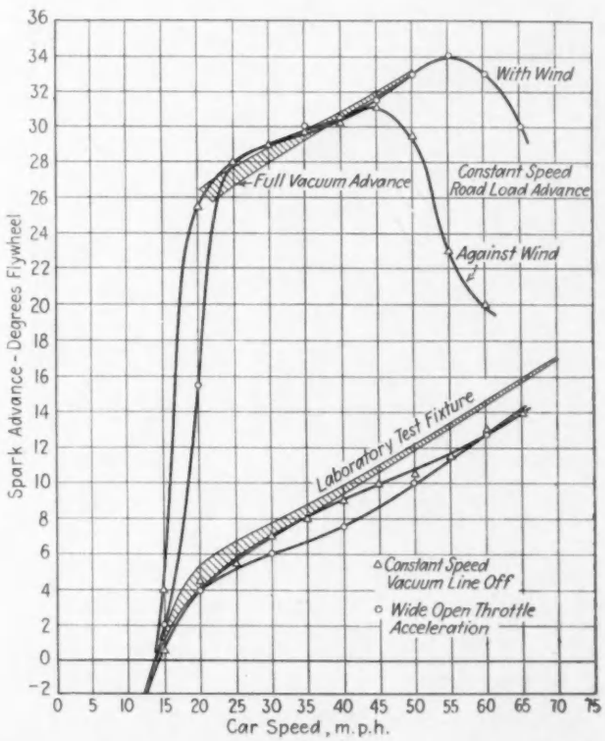

Fig. 7 – Road Spark-Advance Indicator – Recorded from Car No. 2

[1]Fuel *A* is the same as C.F.R. Fuel WA-1; Fuel *B* corresponds to C.F.R. Fuel WB-5; Fuel *C*, C.F.R. Fuel WF-2; and Fuel *D*, C.F.R. WG-1.

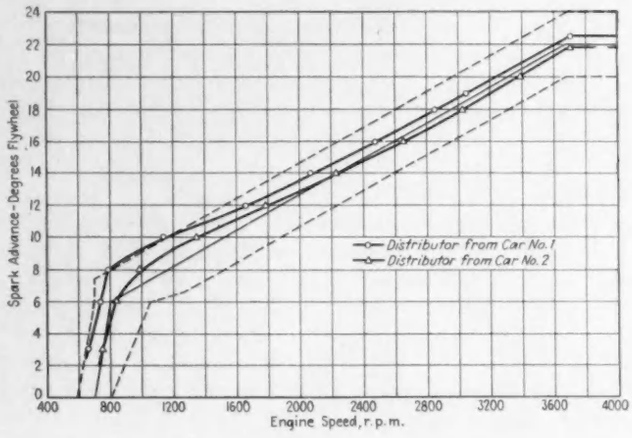

Fig. 8

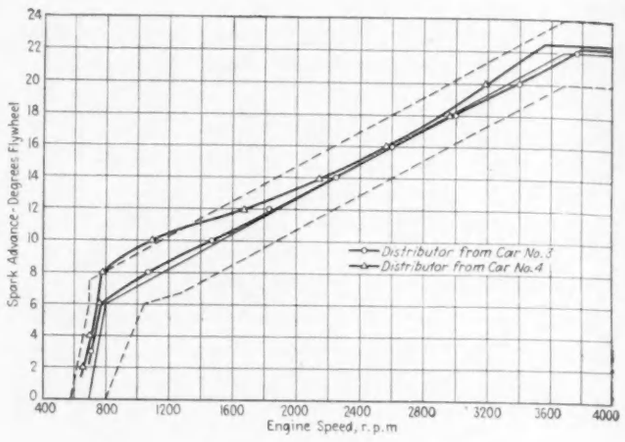

Fig. 9

Figs. 8 and 9 – Distributor Governor Advances – Cars Nos. 1-4

(2) Fuel *B*, the benzol blend, showed a slight but consistent trend towards appreciation as the spark was advanced and knock intensity increased. Greater deviation in ratings obtained on different cars also was observed.

(3) Fuel *D*, the vapor-phase cracked fuel, was depreciated at higher knock intensities, the maximum knock intensity speed changing from 15 m.p.h. to 60 m.p.h. when the distributor was advanced.

(4) A marked variation in knocking characteristics was noted in car No. 4 as compared to the others. Noticeably less sensitive to octane number with an apparent tendency to appreciate the sensitive fuels, this car required an excessive amount of distributor advance to obtain audible knock on these fuels. Car No. 4 was the only privately owned car of the group, its operation probably being much less severe than the company test cars.

The curtailment of the road-rating program, necessitated by the suspension of operations during the strike, has limited the amount of data which were planned originally. We, therefore, are not drawing any definite conclusions based upon these data, but rather are presenting the results obtained in the form of a progress report.

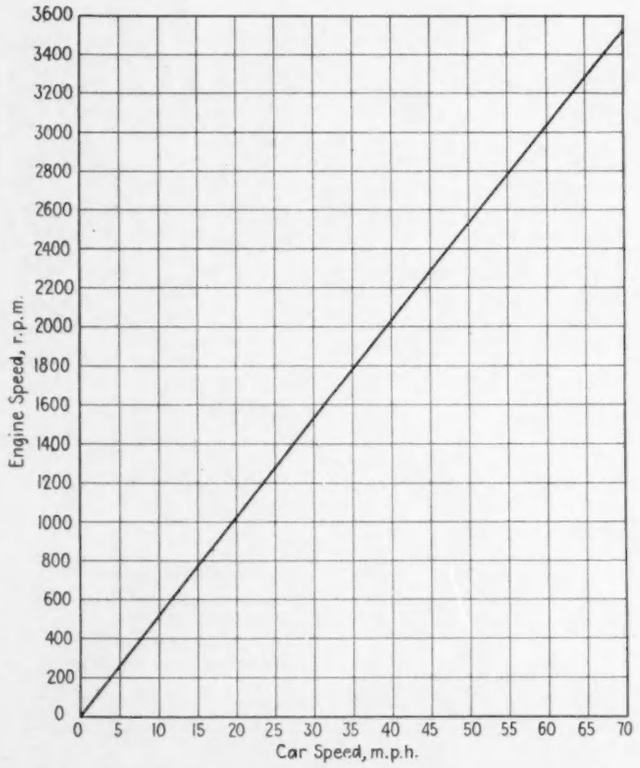

Fig. 10 – Relation of Engine Speed R.P.M. to Car Speed M.P.H. – Cars Nos. 1-4

Results in Non-Changing Motor Oil

THE matter of changing oil in the crankcases of automotive engines is a much-mooted question. The author does not contend that the practice which is being followed is the final answer; however, it is felt that what we have been doing for the last five years is, in our opinion, an economical procedure, if not the most economical, considering all factors. Several years ago, when we were in the throes of the depression we, like all others, were seeking ways and means for effecting economies. Having heard that a local utility had experienced splendid results by replenishing and not changing the oil in the crankcases every 1000 or 1500 miles, we decided to make some experiments on our own account. Accordingly, we discontinued our practice of draining oil every 1000 miles on some 20 Ford cars and simply replenished oil, keeping it at the level recommended by the manufacturer.

Samples of the oil were taken at the termination of each 1000 miles that the cars were driven, and our chemists came to the following conclusions after examining these samples:

Dilution. – This item was found to reach a point of 7 per cent after which it seemed to remain between the limits of 5 per cent and 10 per cent.

Sediment. – It was found that the sediment gradually built up to 1.00 per cent after which, for reasons which we cannot assign, it would – like dilution – remain more or less constant.

Sludge. – The amount of sludge which was found in the crankcases when they were examined seemed to vary considerably but, after rather close scrutiny, it was the consensus of opinion that there was less sludge in the crankcases of the cars on tests than in other cars in which the oil was changed at regular periods.

Acidity. – Although samples were not checked for acidity, we have had but very few cases of bearings being attacked.

Excerpt from the paper of the same title by E. W. Jahn, superintendent of transportation, Consolidated Gas, Electric Light and Power Co. of Baltimore, presented at the Transportation and Maintenance Regional Meeting of the Society, Baltimore, Md., April 16, 1937.

Radio Shielding on Air Transports

By Harold E. Gray

Supervisor, Radio Equipment, American Airlines, Inc.

THE general discussion presented in this paper includes shielded ignition systems, covering both high-tension and low-tension sides of the magneto, generators and generator control boxes, radio power units, motors for general service, and a résumé of airplane shielding and bonding practices.

FOR a long period radio in aviation was somewhat a stepchild – a nuisance to be more or less tolerated, the reason for which no one knew. The part it continued to play and the help afforded by its contribution through the various aids to navigation and flight control gradually put aside the fancied grievances. It was necessary to follow this growth with work in somewhat foreign fields to attain the desired results. One of these departures was to attack the interference problems, foremost of which was that created by the ignition system.

In speaking of radio shielding when associated with aviation, I believe one is generally inclined to think mostly of the shielded ignition harness and spark-plugs although the practice extends to all electrical circuits in the plane. This tendency has no doubt been brought about by the fact that, up to the present time, satisfactory operation of radio equipment as used on aircraft in the United States by transport operators, the Army, and the Navy, has not been obtained without comparatively elaborate shielding on engines.

I believe it sufficient to say that the early attempts were extremely troublesome to all who were in any way brought into contact with them. At the present time few of the old troubles are apparent. The amount of time required to maintain these systems is small compared with that of the earlier types despite the fact that there is a greater need today for more perfect and effective shielding than when first introduced, which systems were the concern of pilot and mechanic alike. The general trend toward better service has brought about developments in the radio field resulting in receivers of very high sensitivity, thus rendering the radio equipment susceptible to extremely small electrical disturbances. The dependence placed on this same equipment in actual flight operations has fostered a critical attitude toward the functioning of a radio set in an airplane, demanding of it the highest performance under the most adverse conditions.

At the present time all shielded ignition harnesses are basically the same. They may, however, be considered as divided into two classes: (1) those employing individually shielded conductors and (2) those using a common shield or manifold from which the individual wires are distributed by means of flexible leads designed to have shielding properties. The termination of the leads at the spark-plugs also differs with the type of spark-plug or spark-plug shield used. Opinions as to the relative efficiency vary both with the manufacturer and user. Comparison is made on mechanical features, weight, and service qualities.

The problem of design as viewed today appears to be more than just one of building a mechanically sound unit that will conduct energy from the magneto to the spark-plug and, at the same time, reduce the radiation of electrical disturbances that so limit the effectiveness of radio operation.

To illustrate this point more fully it will be necessary to depart temporarily from the mechanics involved in the actual construction of the harness and submit for your consideration some of the less apparent features. These features may have relatively little bearing on the effectiveness of the assembly from a purely "radio-shielding" standpoint but, as the development of engines goes on, the influence may be found far-reaching. The presence of electrical capacity in shielded conductors is known, and some efforts have been made to bring about a reduction. However, it is suggested that greater attention be given to this point. In Fig. 1 a group of curves shows the effect on energy available at the spark-plug under various pressures (as tested in a bomb using compressed air) with lengths of individually shielded cables ranging from 36 to 80 in. long as compared with the energy delivered to the plug using an unshielded lead. The desirability of holding the capacity to a minimum is immediately apparent.

A complete ignition harness built of individually shielded conductors, conventional in design and similar to those used regularly on large radial engines, was measured on a test

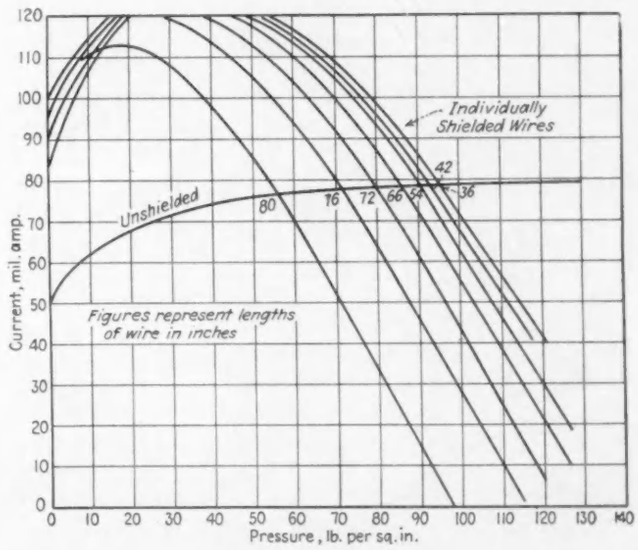

Fig. 1 – Current at Spark-Plug with Various Lengths of Shielded Ignition Wires

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., May 5, 1937.]

Table 1 – Shielded Ignition Harness – Type F Engine
Individually Shielded Conductors

Front

Cylinder	Capacity, microfarads	Atmospheric	Current Measured at Plug 50 lb. per sq. in.	75 lb. per sq. in.	100 lb. per sq. in.	Maximum difference at 100 lb. per sq. in. pressure, 20 per cent.
1	125	58	83	81	80	
2	100	58	88	86	84	
3	110	58	86	84	82	
4	120	58	85	83	81	
5	140	58	80	77	75	
6	190	58	77	73	68	
7	185	58	74	70	67	
8	170	58	77	74	72	
9	145	58	80	77	75	
Plug	0	55	105	102	100	

Rear*

Cylinder	Capacity, microfarads	Atmospheric	Current Measured at Plug 50 lb. per sq. in.	75 lb. per sq. in.	100 lb. per sq. in.	Maximum Difference at 100 lb. per sq. in. pressure. 24 per cent.
1	45	58	90	85	83	
2	140	58	82	78	75	
3	125	58	90	86	80	
4	150	55	80	75	72	
5	170	55	75	72	67	
6	195	50	72	67	63	
7	90	58	87	82	77	
8	80	58	88	84	80	
9	70	58	90	84	80	
Plug	0	55	105	102	100	

(1) Figures are relative and based on value of 100 mil. amp., the value of current obtained with a plug having a lead of no appreciable capacity.
(2) Measurements made with standard aircraft magneto firing plug with 0.015-in. gap. All nine leads of magneto were made common to obtain suitable meter readings.
(3) Values are not absolute.
*High-tension connectors on all rear leads.

bench for capacity and energy being delivered to the various spark-plug terminals under pressures ranging from atmospheric to 100 lb. per sq. in. The results are tabulated in Table 1. It will be observed that at atmospheric pressure there is a comparatively small difference in the values recorded, and that the difference becomes greater as the pressure increases. The decrease in energy is associated closely with the increase in capacity as may be observed in Fig. 2.

To carry the observations further, a standard "manifold"-type of harness employing unshielded wire was tested under similar conditions. The results are tabulated in Table 2. The same general tendency of reduction of available energy at the spark-plug is apparent and of about the same magnitude as in the case of those individually shielded. However, in plotting a curve from these figures as shown in Fig. 3, there

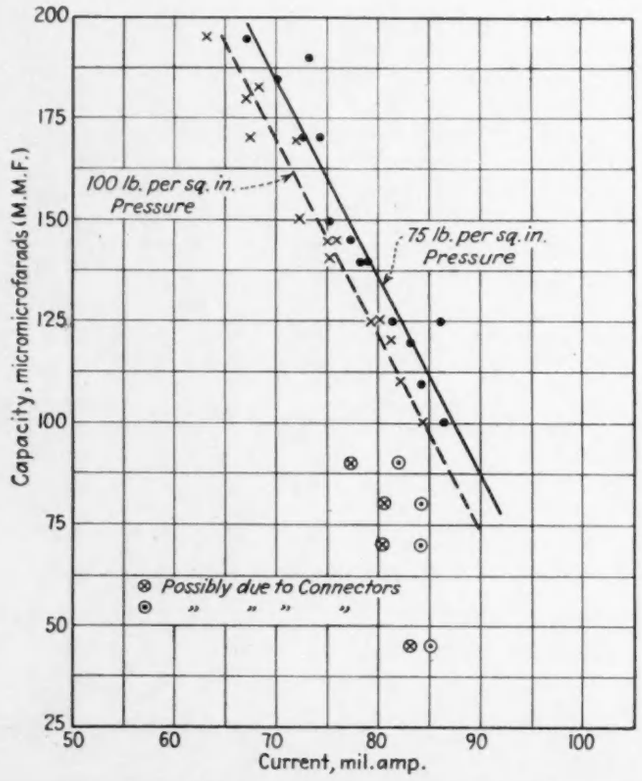

Fig. 2 – Current Vs. Capacity – Shielded Ignition Harness – Type F Engine – Individually Shielded Conductors

is evidence of another force at work which it is suggested might be induction between conductors. This effect could become quite serious as each lead into which energy is induced provides an effective means of draining energy from the one carrying the initial charge. The leakage of energy through these leads probably is increased at the spark-plug itself through spark-plug design, as well as with the age and condition of the plug.

The figures obtained by these tests (Tables 1 and 2) indicate that the same amount of energy is not being delivered to each plug. In the case of the harness constructed of individually shielded conductors there is a difference of 20 per cent between the maximum and minimum energy delivered to the plugs in the front group and a difference of 24 per cent in the rear group when measured at 100 lb. per sq. in. pressure. In the case of the manifold type there appears a difference of 28½ per cent between plugs in the front group and a difference of 25½ per cent in the rear. When one considers the importance of the location of the spark-plug in the combustion-chamber and its influence on engine performance, it is not difficult to surmise that this difference in energy may be an important factor in engine operation.

When running an engine on one magneto there is invariably a drop-off in speed with frequently much less smoothness of operation. Non-uniform ignition is obviously a contributing factor. With a portion of the plugs receiving less energy, a weaker spark is present in those particular cylinders. This spark, which serves as the ignitor, has more work than it is capable of doing, and unquestionably this condition results in some effect on the burning of the charge. In addition, the effect on fuel consumption might be noticeable. If, after reducing operation to one set of plugs and observing the decrease in speed, the throttle was then opened an amount sufficient to restore the original speed there probably would be a change in rate of fuel consumption. Conversely, with full electrical energy being delivered to all cylinders alike, some economies should be obtained. Further, with less energy available at the plug, roughness in the engine can result. This condition may, in a measure, point to the possible cause of some of the innumerable cases of roughness with which we are, for no apparent reason, constantly confronted. When plugs are suspected and removed, bench tests, as frequently as not, show them to be in satisfactory condition. Also, the presence at the plug of a weaker spark, as a result of less energy reaching that point, makes it more susceptible to

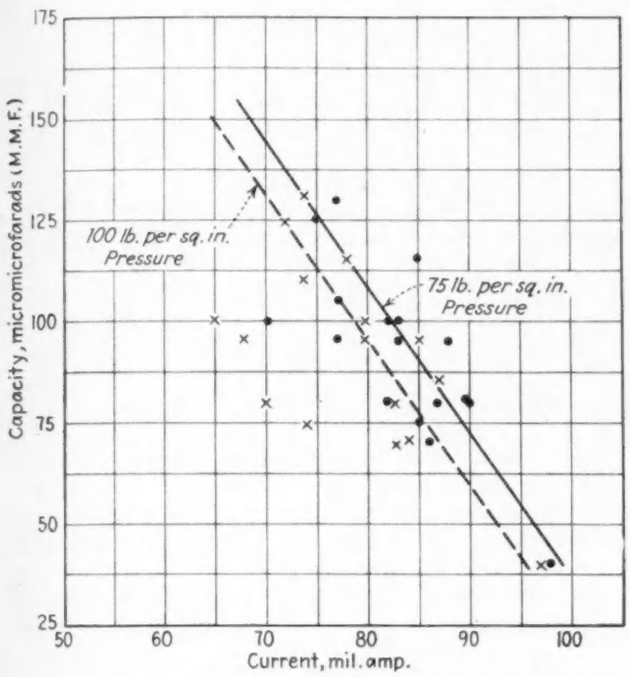

Fig. 3 – Current Vs. Capacity – Shielded Ignition Harness – Type G Engines – Manifold-Type Construction (Unshielded Conductors)

quenching with premature or ultimate fouling of the plug as an aftermath.

Development of higher power engines constantly is calling for larger accessories with greater output. With the presence of losses of much-needed energy, designs are likely to run to sizes and weights greatly in excess of what would be required if the losses could be reduced or eliminated. In addition to a reduction in weight as well as a saving in space, which is always at a premium when one considers accessibility, the saving in power required to drive the larger units would be added to the output of the engine. Although this reduction in itself might be small, when combined with other savings, the increase would surely be acceptable.

The tests made are not necessarily conclusive. It would seem desirable to make them more extensive by having observations taken under more nearly actual operating conditions, such as a motor running on a test stand or even under flight conditions. It is true that some work would be required but, when one considers the effort expended in the development of a modern engine to obtain even the slightest improvements, the work of such an undertaking would not be unreasonable.

No attempt has been made to pass on the type of spark-plug necessary for satisfactory operation in the various engines; however, it has been found that both shielded- and unshielded-type plugs have certain peculiarities that might bear scrutiny.

Due to the construction and method of installation of plugs in an engine, it is not at all unusual that the presence of what might be called "leakage" has not been observed more closely and the magnitude of such "leakage" determined.

For purposes of comparison, the energy passing from the magneto to the plug can be measured by means of a thermocouple galvanometer of suitable range. Ordinarily a measure of the energy to the plug as installed in a cylinder or in a bomb on the test bench by the insertion of a meter in the circuit would be accepted as conventional. However, by isolating the "core" of the plug from "ground" and placing a meter between the external metal portion of the core, which serves to hold it in the shell, and "ground" as illustrated in Fig. 4 somewhat unexpected results are observed.

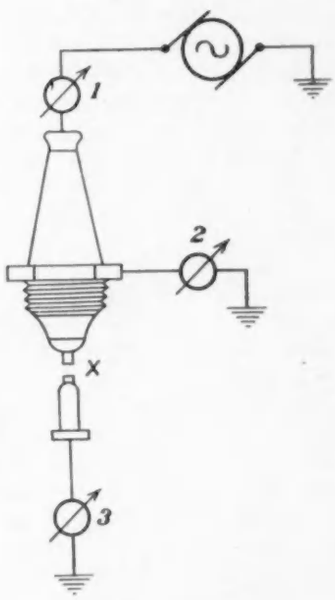

Fig. 4 – Basic Circuit Used for Measurement of "Leakage" Current in Spare Spark-Plug Cores

Table 2 – Shielded Ignition Harness – Type G Engine Manifold-Type Construction (Unshielded Conductors)

Cylinder	Capacity, micro-farads	Atmospheric	Front Current Measured at Plug 50 lb. per sq. in.	75 lb. per sq. in.	100 lb. per sq. in.	Rear Current Measured at Plug Capacity, micro-farads	Atmospheric	50 lb. per sq. in.	75 lb. per sq. in.	100 lb. per sq. in.
1	70	63	94	86	83	100	59	87	83	80
2	70	67	97	89	84	100	59	84	82	80
3	80	63	92	87	83	125	55	77	75	72
4	95	63	89	83	80	130	59	80	77	74
5	95	63	83	77	68	115	61	88	85	83
6	105	64	83	77	74	95	63	92	88	85
7	100	64	84	70	60	85	60	94	90	87
8	80	64	95	82	70	40	63	100	98	97
9	75	67	95	85	74	85	63	95	90	87
Plug	0	55	105	102	100	0	55	105	102	100

Maximum difference at 100 lb. per sq. in. pressure, 28½ per cent.

Maximum difference at 100 lb. per sq. in. pressure, 25½ per cent.

(1) Figures are relative and based on value of 100 mil. amp., the value of current obtained with a plug having a lead of no appreciable capacity.
(2) Measurements made with standard aircraft magnetc firing plug with 0.015-in. gap. All nine leads of magneto were made common to obtain suitable meter readings.
(3) Values are not absolute.

With current flowing across the gap at *X*, a considerable amount of energy is observed to pass through meter 2. This energy may be the result of the capacity existing between the spindle and the metal portion of the core. On some types of plugs tests conducted at atmospheric pressure disclosed as high as 30 to 35 mil. amp. current being recorded on meter 2 with approximately 100 mil. amp. showing on meter 1. On other types the current in meter 2 indicated approximately 20 mil. amp. with meter 1 showing in excess of 125 mil. amp. These values show a ratio of approximately 3:1 in the first case, and approximately 6:1 in the latter. In all cases a standard aircraft magneto was used as a source of power. The "leakage" current observed on porcelain plugs, in general, was less than that observed on mica plugs passing the same amount of current. Both domestic and foreign plugs were tested and seemed to have about the same characteristics.

Exhaustive tests were not made to determine what influence the presence of this "leakage" current might have on spark-plug performance but, in consideration of the fact that plugs passing greater currents apparently had less "leakage" losses, it would seem that this feature could well be considered in plug design with a view toward a further contribution to the more effective use of the ignition energy.

The mechanics of ignition-harness construction are by no means devoid of complications. With the use of high-output engines the problem of heat was introduced. Wires adjacent to the spark-plugs burned badly, causing the insulation to rupture. Both radiated and conducted heat seemed to be present. Compounds used for years as suitable insulation were found to be inadequate.

Various manufacturers were called upon to supply a more heat-resisting product. New types of spark-plug connectors were introduced, some with provisions for large air spaces between the shield and the conductor, others with solid insulation. The use of connectors with solid insulation based on actual tests seemed to be the most desirable method of approach, but development along these lines was hampered due to the limited number of materials available that would be capable of resisting the high temperatures, that would lend themselves to practical manufacturing processes, and that would have sufficient durability to warrant their adoption. It was found that protection against this burning was only required for a distance of about 3 in. back from the plug connector, except for radiation effects produced by the proximity of the exhaust system. There is still need for such an accessory.

Changes were made in the method of distributing wires from the ignition manifold to the cylinders, to provide better cooling. Each of these points contributed a slight improvement until a point was reached where reliable operation was obtained. However, it is felt that we are not far removed from this problem particularly with engines of higher ratings and complexity of cylinder arrangement.

The presence of moisture, particularly at the connectors, has contributed more than its share of trouble. The entrance of this moisture was found to be the result of both improperly sealed connections and condensation. It also was determined on certain types of ignition-wire construction that the layer of cotton braid introduced directly over the rubber insulation was acting as a wick. Upon becoming saturated a leakage patch was provided, apparently of lower resistance than the gap of the plug itself. The spark was found to travel along this saturated braid underneath the outer covering of the wire and jump through the thin layer of outer braid and lacquer to "ground" some ¾ in. behind the terminal. Correction was provided by treating the ends of all wires employing this type of construction with an insulating varnish.

Small copper gaskets were introduced between the outer end of the spark-plug and the connector and suitable rubber gaskets at the point where the connectors fastened to the lead running from the manifold. In the use of rubber gaskets more attention could be paid to their design. There has been a tendency to use gaskets that do not permit complete seating of the coupling nuts resulting in less effective bonding. In addition, as the coupling nuts are tightened, excess pressure is exerted by the gasket on the ignition wire creating a stricture in the insulation as a result of cold flow. Cases have been found where the wire was practically cut in two. In nearly all cases the diameter of the insulation at these points is reduced to half. There is an eventual rupture due to the high voltage being carried. To control this tendency the use of rectangular gaskets seated in a form of packing gland have been found superior to a tapered gasket in a tapered seat.

With the features of heat resistance and moisture exclusion taken care of, there still remains a problem of maintaining good mechanical connections that will afford low electrical resistance in the shield. As there are so many points of contact, this feature requires considerable attention. In the average manifold type of harness there are 42 such joints per engine exclusive of the magneto shields. A poor electrical connection at any one of these joints is a potential source of interference. It is important that they remain tight from the time of installation until the overhaul period is reached. If they become loosened, oil and dirt find their way on to the contact surfaces and, although the mechanic may tighten them, he seldom takes the trouble to clean them thoroughly beforehand. In many cases the joints are not readily accessible, hampering maintenance to no small degree. It has been necessary to provide special locking devices at the less-accessible locations to insure against joints coming apart. All of this preparation must be done so that routine maintenance does not become more involved.

In general, the maintenance of a harness installed on the front of an engine is less difficult than one installed on the rear. However, such an installation is subject to greater wear and tear as a result of constant work around the engine.

Magneto shields have been another great source of trouble. Adequate means have not been provided in the past to prevent the entrance of oil and water. The contact surfaces have been insufficient and the general design such as to make the installation or removal from the magneto while on engines, extremely difficult. Considerable attention could be given to improved magneto-shield design.

Off hand one might be inclined to overlook the "ground" wires running from the magnetos, giving little thought to the presence of energy in them. Interference from the low-tension side of the magneto is as effective in limiting radio operation as that produced by the high-tension side. The wires from the magneto to the ignition switch must be shielded carefully. It is undesirable to run these wires in the same conduit with battery wires as the energy present in them can be induced in adjacent circuits and carried to all parts of the ship.

Aside from shielding the ignition system on an airplane, a great deal of attention is devoted to general bonding and shielding. Practically all electrical circuits are carried in conduit. Electrical panels are enclosed in metal boxes. Control surfaces are bonded across the hinge points. Control cables are bonded at points of attachments and around turnbuckles. In general an attempt is made to tie all metal parts together electrically. Although a metal member or control wire may be bonded at each end, if any appreciable length exists between bonds, noise may result if the part comes in contact with other metal parts as there may be a difference in voltage along its length due to charges accumulating, and the

(*Continued on page* 544)

High-Pressure Viscosity as an Explanation of Apparent Oiliness

By H. A. Everett
The Pennsylvania State College

OILS from three different crudes, matched as to initial viscosity but with widely different viscosity indexes, gave markedly different values when tested in an oiliness machine. As viscosity index indicates temperature effects only, the effect of pressure on viscosity was investigated in a high-pressure viscometer. For each oil, values up to 50,000 lb. per sq. in. pressure were obtained and for three temperatures. From these data characteristic curves were plotted giving complete pressure, viscosity, and temperature (P.Z.T.) relations. Using such curves it was possible to trace the changes in viscosity which each oil underwent in its passage through the bearing and obtain an estimate for equivalent viscosity. When this investigation was made the different so-called "oiliness" effects were shown to be but the normal effect of the true viscosity actually existing in the oil film.

The paper calls attention to the need for information on the influence of pressure on viscosity as well as of temperature on viscosity when comparing or predicting the performance of oils in service bearings.

OILINESS, as the term has been used for the past several decades, has not been defined rigidly, nor have the devices built for its measurement been standardized, nor has any standard technique been developed for its evaluation. Herschel's definition of oiliness[1] specifies "the property that causes a difference in the friction when two lubricants of the same viscosity at the temperature of the film are used under identical conditions." In harmony with this definition, several machines have been in use which measure relative friction when supplied with oils of matched viscosities, thus ostensibly classifying oils as to relative oiliness.

For the past several years there has been in use in our lubrication laboratory a machine developed by Prof. L. J. Bradford, operating on the foregoing principle[2]. All oils used in engine tests have been tested in this machine as a routine matter, and it is with the vagaries of performance which the oils show when passing through the test bearing of this machine that this paper deals.

The machine, shown diagrammatically in Fig. 1, consists essentially of a small fitted bearing of brass, *b*, (0.135 x 0.271 in.) supported on a knife edge and pressed upward against a journal, *a*, (1⅞ in. diameter). The bearing is free to tilt so that the necessary wedge-shaped film convergent in the direction of motion can be formed, and may be loaded as desired. Just below the surface (1/32 in.) of the bearing is located a thermocouple, and another thermocouple is located external to the bearing in the stream of oil supplied, *c*. These couples give temperatures approximating those existing in the oil film within the bearing and in the oil prior to entering the bearing. The journal is made from the outer race of a tapered roller bearing (1⅞ in. diameter) and has a surface meticulously smoothed and mated to the bearing. The journal is rotated at constant speed, (465 r.p.m.), and the bearing is loaded progressively by 500-lb. increments. The difference in temperature between that of the entering oil and the thermocouple junction bedded in the bearing is considered as indicative of the amount of work done within the oil film in passing through the bearing. Accordingly, the graph of this temperature difference, termed ΔT, plotted as a function of bearing load, establishes the relationship between the internal friction of the oil film and the bearing load. Provided that the surface of the journal and bearing are fitted accurately to each other and of extreme smoothness, no difficulty is experienced in obtaining mean bearing loads up to 30,000 lb. per sq. in., and repetitive results can be obtained readily. This machine has been described more fully in an article appearing in the January, 1937, issue of *Machine Design*, by Bradford and Wetmiller[3].

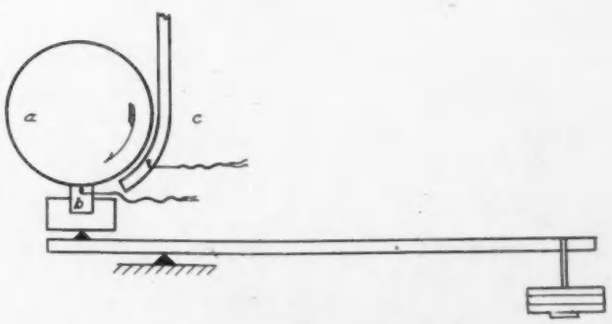

Fig. 1 – Experimental Set-Up

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., May 6, 1937.]

[1] See S.A.E. TRANSACTIONS, 1922, Vol. 17, Part I, pp. 282-320: "Viscosity and Friction," by W. H. Herschel.

[2] See Bulletin 44, The Pennsylvania State College Engineering Experiment Station, pp. 20-21: "Performance Tests of Lubricating Oils in Automobile Engines," by H. A. Everett and F. C. Stewart.

[3] See *Machine Design*, January, 1937, pp. 36-39: "Attaining the Maximum in Bearing Performance," by L. J. Bradford and R. S. Wetmiller.

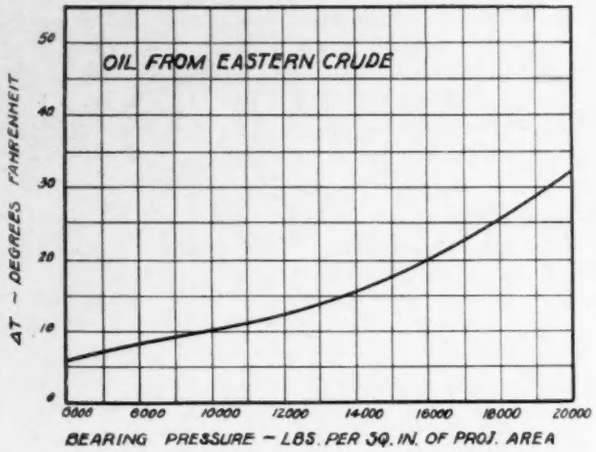

Fig. 2 – Typical Curve of ΔT Vs. Load as Obtained in the Experimental Bearing

A typical curve for an eastern oil is shown in Fig. 2. When matched as to viscosity, oils from the same crude but of different blends gave practically the same curve. However, oils produced from crudes of different fields gave decidedly different curves. At the same bearing load the ΔT for one oil might be as much as twice that of an oil from another field. To explain such wide divergences as due to differences in "oiliness" seemed unreasonable. It was, of course, recognized that the viscosity decreased with rise in temperature and also that oils with different viscosity-temperature relations (or viscosity indexes)[4] would have correspondingly different viscosities at the temperature of the bearing even though matched at the supply temperature. However, oils which had notably lower viscosity indexes and, therefore, at bearing temperature had lower viscosities, did not give the lower values of ΔT but conversely gave the larger values.

A series of tests was made using three oils all matched as to viscosity at 130 deg. fahr. but from western, mid-continent, and eastern crudes. The physical inspection data are given in Table 1. The A.S.T.M. viscosity-temperature graphs are shown in Fig. 3 and from them it is obvious that the oil from the western crude had a viscosity at bearing temperature,

[4] See *Chemical and Metallurgical Engineering*, October, 1929, p. 618: "Viscosity Variations of Oil with Temperature," by E. W. Dean and A. H. B. Davis.

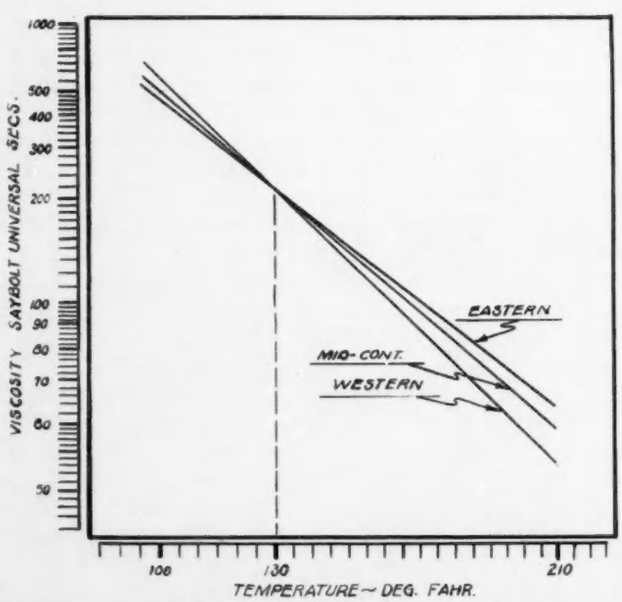

Fig. 3 – A.S.T.M. Viscosity-Temperature Graphs of the Three Oils Used

Table 1 – Physical Properties of the Oils

	Eastern	Western	Mid-Continent
Gravity, A.P.I., deg.	28.6	20.1	25.1
Specific Gravity at 60 deg. fahr.	0.8841	0.9334	0.9036
Saybolt Viscosity at 100 deg. fahr., sec.	460	582	500
Saybolt Viscosity at 130 deg. fahr., sec.	212	218.5	214
Saybolt Viscosity at 210 deg. fahr., sec.	61.6	53.8	58.7
Flash point, deg. fahr.	450	395	450
Fire point, deg. fahr.	505	445	505
Pour point, deg. fahr.	30	—5	0
Color, A.S.T.M.	7	3.5	6
Conradson Carbon, per cent	0.60	0.14	0.49
Steam Emulsion No.	240	270	260
Viscosity Index	103	—20	74
Gravity Index	101	17	79

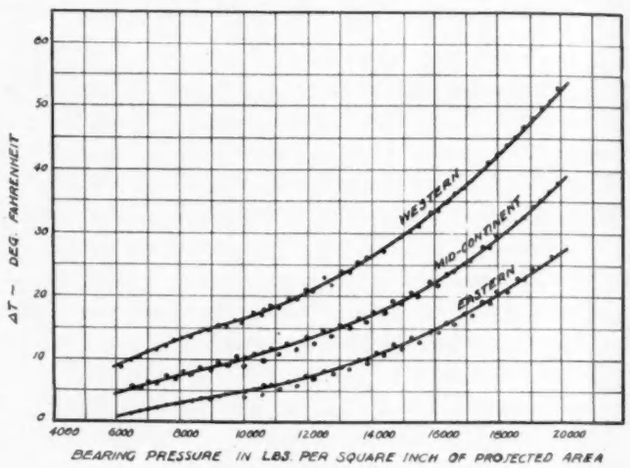

Fig. 4 – Curves of Temperature Rise (ΔT) Vs. Load as Obtained for the Three Oils

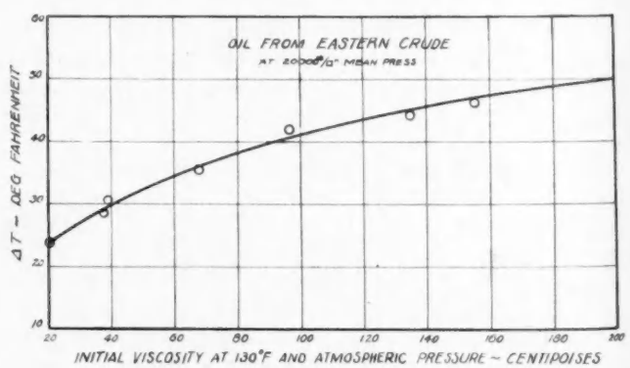

Fig. 5 – Effect on ΔT (Resistance) of Change in Viscosity (All Oils Had the Same Viscosity Index)

materially lower than either of the others. The results from the Bradford machine (ΔT vs. bearing pressure) are shown by the curves of Fig. 4.

Considering the 20,000 lb. per sq. in. loading as the basis of comparison, the values of ΔT as observed were 27, 38 and 53 deg. fahr. for the eastern, mid-continent, and western oils, respectively. Corrections to allow for heat losses from the bearing due to conduction, radiation, and so on, were estimated at 4, 5, and 11 deg. fahr. respectively which brought the corrected values to 31, 43, and 64 deg. fahr. The resistance of the oil film was considered proportional, or nearly proportional, to ΔT, as the internal work done on the film could only be dissipated in heating the oil to its outgoing temperature, hence the resistances actually offered by the films of these three oils were of the order of the corrected ΔTs or 31, 43, and 64 deg. fahr. for the eastern, mid-conti-

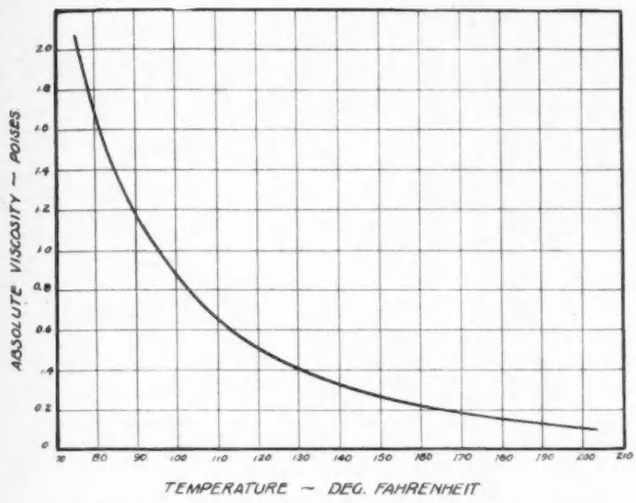

Fig. 6 – Typical Curve Showing the Effect of Temperature on Viscosity with Pressure Retained Constant (Atmospheric) – Oil from Eastern Crude

nent and western oils respectively. Considering only the effect of temperature, the western oil would be at the lowest viscosity of all three when passing through the bearing, hence the oil of least viscosity was giving the highest resistance.

According to orthodox theory the resistance offered by a lubricating film should be a function of the viscosity and an inverse function of the film thickness. For parallel plates the resistance $=$ (area) $\frac{\text{(velocity) (viscosity)}}{\text{(film thickness)}}$. For convergent films, such as existed in this bearing, the expression is more complex involving the geometry of the film in addition to the factors of velocity and viscosity[5]. As the minimum film thickness changes in a manner not directly proportional to the viscosity, the effect on resistance (or ΔT) of a change in viscosity could not be predicted off-hand. Accordingly, recourse was had to experiment to determine the effect that variation in viscosity alone would have on ΔT. Runs were made with a series of oils having the same viscosity index but varying viscosities. The results are shown in Fig. 5 in which, for a constant mean pressure of 20,000 lb. per sq. in., the values of ΔT are plotted for oils whose viscosities, at 130 deg. fahr. and atmospheric pressure, varied from 20 to 160 centipoises. From this curve it is apparent that higher viscosities give larger values of ΔT.

In view of this relation, the experimental fact that the oil with the lowest viscosity index gave the highest ΔT indicated that the change produced in viscosity by pressure as well as by temperature must be reckoned with. A rise in temperature causes a decrease in viscosity (see Fig. 6), whereas a rise in pressure causes an increase in viscosity (Fig. 7). At low pressures the temperature effect might predominate whereas, at high pressure, the rise in viscosity due to pressure might overshadow the drop in viscosity due to temperature. Convinced that data must be obtained which would show the effect of pressure on viscosity, there was established in the School of Chemistry and Physics of The Pennsylvania State College, a high-pressure laboratory under the immediate supervision of Dr. R. B. Dow.

[5] See A.S.M.E. Transactions, 1932, RP-54-7, pp. 123-148: "Optimum Conditions in Journal Bearings," by A. Kingsbury.

[6] See "The Effect of Temperature and Pressure on the Viscosity of Lubricating Oils," by R. B. Dow, presented at The New York Meeting of the Society of Rheology, October, 1936, printed by the American Institute of Physics, New York City.

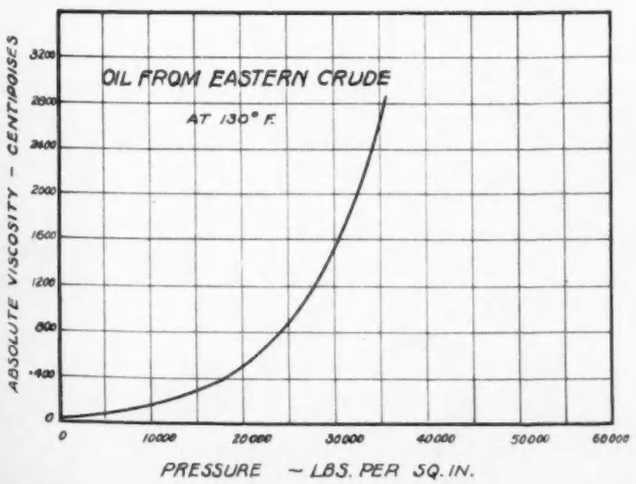

Fig. 7 – Effect of Pressure on Viscosity with Temperature Retained Constant (130 Deg. Fahr.)

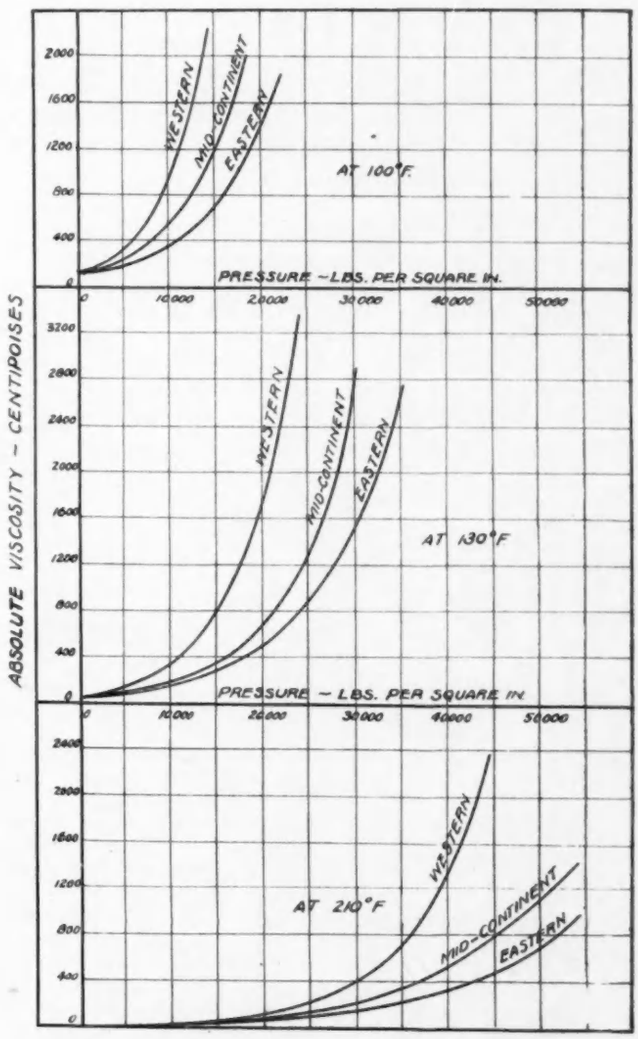

Fig. 8 – High-Pressure Viscosities for Three Oils, Each from a Different Crude at Three Different Temperatures

In this laboratory the viscosities at various pressures up to 50,000 lb. per sq. in. were determined for three temperatures: 100 deg. fahr., 130 deg. fahr., and 210 deg. fahr. These data were obtained for the three oils of Fig. 4 and are tabulated in Table 2 taken from Dow's Report[6]. They are shown graphically in Fig. 8. As the viscosity-pressure relations were obtained for three temperatures, it was possible to plot cross-curves which show the effect on viscosity of both pressure and temperature. Such characteristic curves combining the properties of pressure, viscosity, and temperature (P.Z.T.)

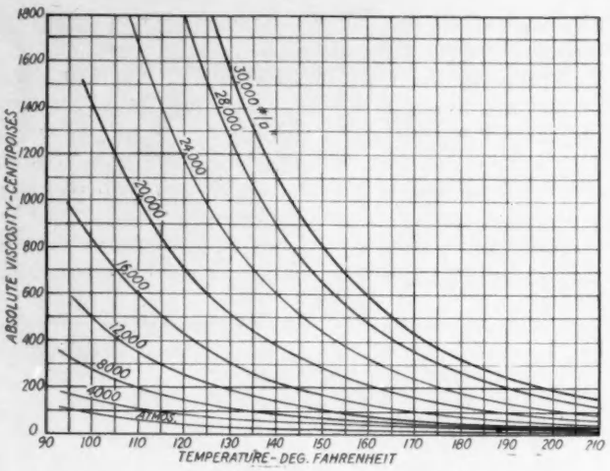

Fig. 9 – Skeleton P-Z-T Chart for Oil from Eastern Crude

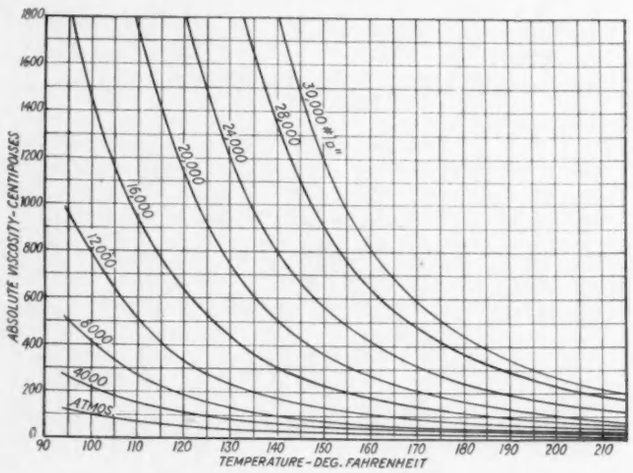

Fig. 10 – Skeleton P-Z-T Chart for Oil from Mid-Continent Crude

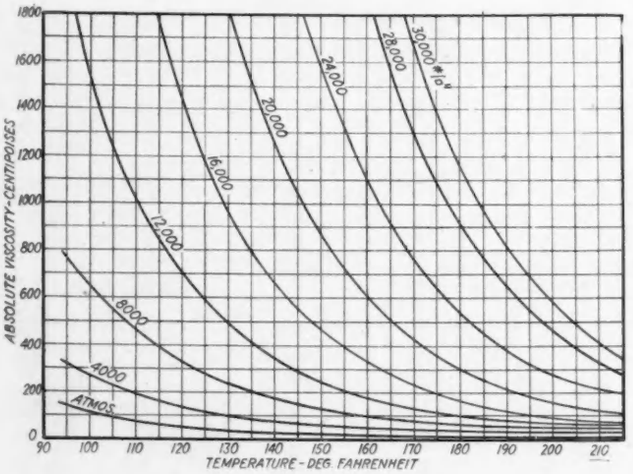

Fig. 11 – Skeleton P-Z-T Chart for Oil from Western Crude

are shown in skeletonized form in Figs. 9, 10, and 11. From such charts, for any combination of pressure and temperature, the corresponding viscosity may be read directly.

Such comprehensive charts make it possible to obtain an idea of the successive viscosity changes experienced by a particle of oil in its travel through the bearing. For example, the oil droplet reaches the bearing at a temperature of 130 deg. fahr. and at atmospheric pressure; then, shortly after entering the bearing, the pressure rises to a maximum accompanied by a corresponding temperature rise. As it approaches

Table 2 –
Viscosity-Pressure Data As Determined by R. B. Dow, 1936[a]

Pressure, lb. per sq. in.	Viscosity, centipoises: Pennsylvania	Oklahoma	California
100 Deg. Fahr.			
14.2	83 86*	94 97*	114 116*
1000	91	119	146
2000	106	145	183
3000	123	175	225
4000	145	209	278
5000	169	247	346
6000	198	293	433
7000	232	344	533
8000	268	405	655
9000	310	475	811
10×10^3	357	557	995
12	485	775	1540
14	654	1060	2200
16	850	1430	
18	1100	1940	
20	1420		
22	1830		
130 Deg. Fahr.			
14.2	41 39*	43 41*	42 43*
1000	51	54	57
2000	60	66	68
3000	64	75	80
4000	73	85	99
5000	82	95	124
6000	91	102	154
7000	100	118	190
8000	111	131	232
9000	124	149	281
10×10^3	143	170	340
12	191	231	490
14	249	318	692
16	315	428	960
18	408	564	1320
20	524	740	1830
22	663	940	2510
24	830	1170	3400
26	1030	1500	4540
28	1260	2030	
30	1660	2840	
32	1960		
34	2460		
210.2 Deg. Fahr.			
14.2	7 9*	10 9*	13 8*
1000	9	12	15
2000	10	14	17
3000	13	16	19
4000	15	18	21
5000	17	20	24
6000	19	22	25
7000	20	24	27
8000	22	27	30
9000	24	30	33
10×10^3	26	33	37
12	31	40	45
14	36	50	55
16	44	59	70
18	52	71	90
20	62	85	116
22	73	104	153
24	87	128	202
26	103	154	260
28	123	185	327
30	145	219	408
32	171	262	510
34	202	314	655
36	242	375	846
38	387	454	1080
40	337	549	1400
42	393	654	1790
44	457	770	2270
46	535	890	2890
48	627	1010	
50	732	1150	
52	846	1290	
54	973	1430	

*As revised by Petroleum Laboratory.

the exit edge, the pressure grows less but the temperature would continue to rise until, at exit, the temperature would be at a maximum, whereas the pressure would return to atmospheric as the droplet left the bearing. All of these changes can be traced on the P.Z.T. charts.

If the exact manner in which the pressure and temperature varied throughout the bearing film were known, it would be possible to trace on Figs. 9, 10, and 11, the viscosity history of any particle of oil from the time it entered the bearing until it left. Lacking such exact information one can only make certain assumptions as to the character of the path of travel. Such assumptions may be justified by convenience or basic theory; nevertheless, they are assumptions and, of necessity, vitiate quantitative comparisons. However, even though the results may have little quantitative value, it is interesting to follow the private life of the oil particle as made possible by these simplifying assumptions. For example, if one assumes the film to have a uniform pressure throughout the bearing equal to the mean pressure, the cycle is quite simple. For such the pressure and viscosity on entering the bearing would immediately rise (line *A-B* in Fig. 12) to the value of the mean pressure (20,000 lb. per sq. in.) after which the pressure would remain constant but the temperature would increase and the viscosity decrease (line *B-C*) until the exit was reached. In Fig. 12 the line *A-B-C-D* thus indicates such successive changes from entrance to exit. Such a highly conventionalized cycle is undoubtedly far from the true picture; yet it treats each oil alike, and the differences due to the viscosity characteristics of the different oils are striking.

In Fig. 13 these simplified cycles for each of the oils under consideration are shown superposed. It is interesting to note that, although the terminal viscosities (at *C*, *C'* and *C''*) just prior to exit, are not greatly different, the viscosities experienced in transit differ enormously. For example, a particle of the eastern oil reaches a maximum viscosity at (*B*) of approximately 525 centipoises, the mid-continent at (*B'*) 740 centipoises, and the western at (*B''*) 1830 centipoises, whereas all have viscosities at exit of around 200 centipoises. In more detail, the data are as follows:

	Eastern	Mid-Continent	Western
Viscosity at entrance (Atmospheric Pressure), centipoises	40	40	40
Viscosity after entrance, centipoises	525	740	1,830
Viscosity just before exit, centipoises	200	180	190
Viscosity just after leaving (Atmospheric Pressure), centipoises	20	15	15

The assumption of a uniform pressure throughout the bearing, on which the foregoing was based, is so widely divergent from what actually occurs that it might almost be considered fantastic were it not that it utilizes the only experimentally observed factor (mean pressure) and permits comparisons to be made readily. Actually the pressure in the oil film varies circumferentially from zero at entrance to a maximum somewhere near the center of the bearing and falls again to zero at the trailing edge. Moreover the axial distribution also varies from zero at the bearing edges to a maximum near the center of the axial length. Thus the pressures existing over the bearing surface, if plotted to scale, would form a somewhat tent- or dome-shaped figure rising to a peak near the center but becoming zero around the edges. For each of these pressures, there is a corresponding temperature and viscosity. Were these values known it might be possible to obtain one viscosity, equivalent in its effect to the many viscosities actually existent. Unfortunately, only mean

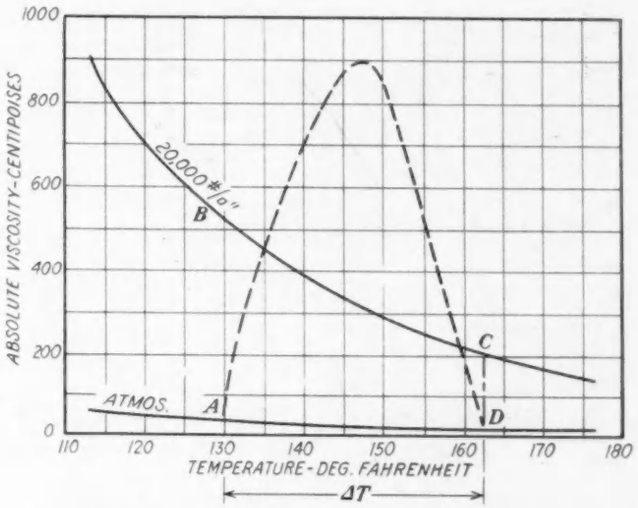

Fig. 12 – Progressive Viscosity Changes Experienced by an Oil Droplet in Passing Through the Bearing – Droplet Before Entering is at State *A* and, After Leaving, Is at State *D*

values were available for pressures and temperatures; therefore, reliable quantitative comparisons could not be made.

A cycle more closely approximating actual conditions can be sketched if one is willing to assume a maximum pressure in terms of mean pressure and a path for its attainment. Such a cycle is shown by the dotted line of Fig. 12 and, in detail, for each oil in Figs. 14, 15, and 16. For these curves a maximum pressure (30,000 lb. per sq. in.) was taken equal to 1.5 of the mean pressure (20,000 lb. per sq. in.), and the only guide in selecting the path to and from this peak was that the temperature increments should be higher at the higher pressures when considering equal pressure steps. The mean

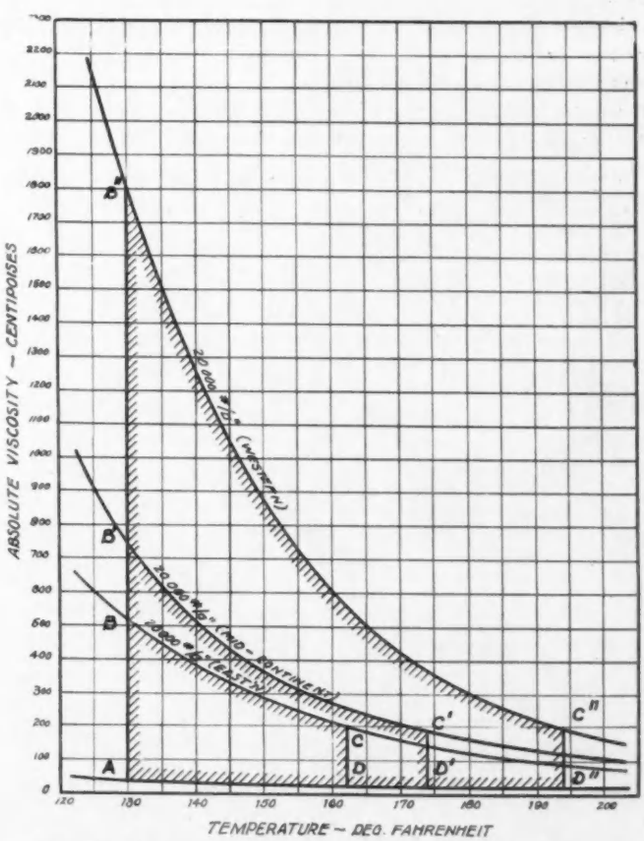

Fig. 13 – Simplified Cycles for the Three Oils Shown Superposed – All for Bearing Loaded to 20,000 Lb. per Sq. In. Mean Pressure

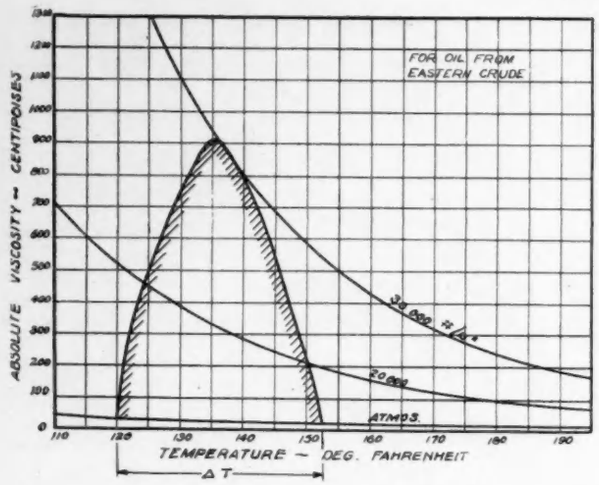

Fig. 14 – Viscosity Changes Experienced by an Oil Particle When Passing Through the Bearing, Allowance Being Made for the Non-Uniform Pressure Existing in the Oil Film – Oil from Eastern Crude

height of the dome might be taken at two-thirds of the maximum, hence the viscosity at this point (and at mid-point of ΔT) might be considered the equivalent viscosity or a single viscosity equivalent to the variable viscosity throughout the film.

Values for equivalent viscosities obtained in this manner are 610, 730, and 1470 centipoises for the eastern, mid-continent and western oils respectively. From these values it is evident that the operative viscosity of the oil film is materially higher for the western and mid-continent oils than for the eastern oil, thus in harmony at least qualitatively with the values of ΔT experimentally obtained.

As mentioned earlier the resistance offered by a film in a bearing is affected by both viscosity and film thickness. Film thickness, for a constant load, is in turn influenced by viscosity, therefore, by both temperature and pressure. The effect of these two factors (temperature and pressure) on film thickness considered separately and combined as computed for a bearing of this type[7] when subjected to progressively

[7] Computed by Reynolds' equation for convergent bearing; see Reference 3 for details.

[8] Reynolds' equation for the resistance offered by the film in a convergent bearing is:

$$\text{Resistance} = \frac{(\text{Viscosity})\ (\text{Velocity})}{(\text{Angle of Convergence})} \left[4 \log_e \frac{b}{a} - \frac{6\,(b-a)}{(b+a)}\right]$$

where a and b are functions of the geometry of the bearing; for details see Reference 3.

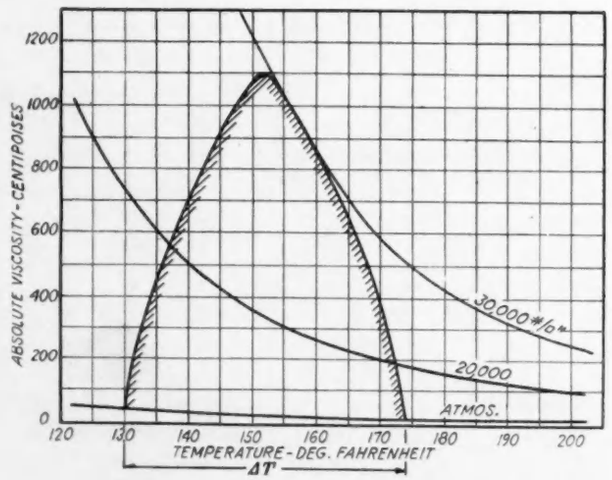

Fig. 15 – Viscosity Changes Experienced by an Oil Particle When Passing Through the Bearing, Allowance Being Made for the Non-Uniform Pressure Existing in the Oil Film – Oil from Mid-Continent Crude

increasing loads is shown in Fig. 17 reproduced from the article by Bradford and Wetmiller[3].

From this figure the complex variation of film thickness consonant with rising pressure and temperature as encountered in an actual bearing of this type is apparent and the futility of offhand predictions obvious. However, with equivalent viscosities known, computations become possible for film thickness, resistance, and work all in accordance with accepted hydrodynamic theory[8]. Accepting the earlier mentioned equivalent viscosities as estimated for the 20,000 lb. per sq. in. loading, the following values were computed:

	Eastern	Mid-Continent	Western
Minimum film thickness, in.	5.52×10^{-5}	6.03×10^{-5}	8.55×10^{-5}
Resistance, lb.	2	2.2	3.1
Work, in-lb. per sec.	94	104	146

The correlation between these computed values of the

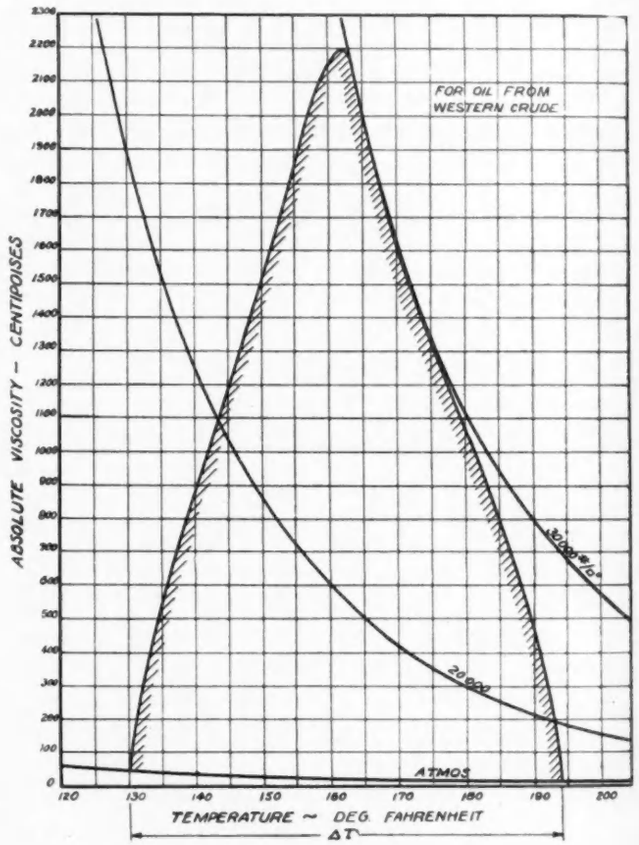

Fig. 16 – Viscosity Changes Experienced by an Oil Particle When Passing Through the Bearing, Allowance Being Made for the Non-Uniform Pressure Existing in the Oil Film – Oil from Western Crude

work to overcome friction, with the experimentally determined ΔTs, (31, 43, and 64 deg. fahr. respectively) is interesting and significant. Qualitatively it is all that could be desired; quantitatively it is a bit disappointing though probably a closer agreement should not have been expected. The discrepancy can be attributed to several factors: the difficulty of establishing corrections to the observed ΔTs to allow for heat losses due to conduction, radiation, and so on; the method by which the value of an equivalent viscosity was determined; the crudeness of the determination of the film temperature, and so on.

Notwithstanding all these apologia there has been a sincere

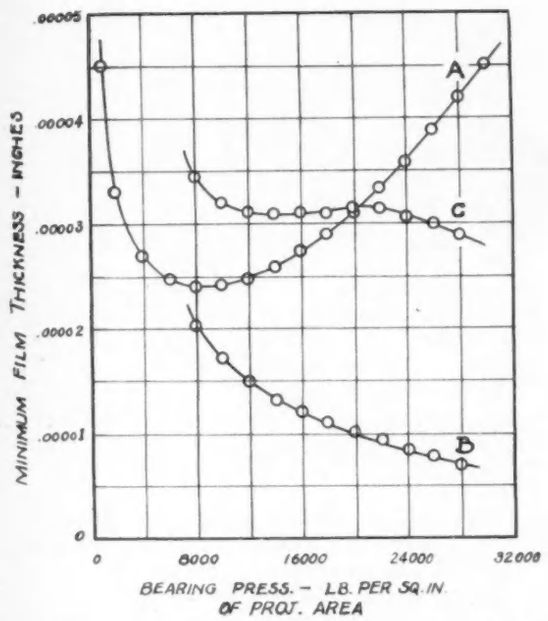

Fig. 17 – Effect of Temperature and Pressure on Computed Film Thickness for Convergent Bearing – Subjected to Progressively Increasing Loads

Curve *A* shows effect of pressure alone.
Curve *B* shows effect of temperature alone.
Curve *C* shows effect of temperature and pressure combined.

attempt to take cognizance of the effects of variation in both pressure and temperature on the viscosity throughout the oil film and to explain some phenomena attributed to "oiliness" as mainly the result of viscosity changes. The high-pressure data show so clearly the great differences that exist in different oils that it seems obvious that data of this sort are as essential as the temperature data when considering the performance of an oil in a bearing. Without the P.Z.T. charts or their equivalent we cannot explain or predict oil-film performance in bearings.

In closing I wish to make it quite clear that I am not presenting a carefully considered theory or finished solution, nor even completed experimental work, simply some experimental facts which, at first glance, are apparently out of step with our concept of viscosity but not really so when analyzed with the aid of more complete information.

Credit for the laboratory data is due to R. S. Wetmiller, research assistant in our lubricating laboratory under the immediate supervision of Prof. L. J. Bradford. Both have been most willing co-workers in preparing the material for this paper.

Discussion

Questions Validity of Equations Used

– H. C. Dickinson

Chief, Heat and Power Division, National Bureau of Standards

IN his discussion of Professor Everett's paper, published herewith, Professor Lay outlines briefly the nature of viscosity. I should like to extend some of this elementary discussion to the ordinary shaft or connecting-rod bearing.

If we have two parallel planes *A* and *B*, as Professor Lay pointed out, separated by a layer of oil, as shown in Fig. *A*, the basic assumption, borne out by experience, is that the force required to slide the planes relative to each other so as to shear the oil layer is proportional to the area of the planes, to the speed at which they move, and inversely proportional to the distance between the sliding surfaces, provided the space is uniformly filled with the lubricant. This relationship assumes, of course, that the oil is of uniform viscosity and incidentally requires that it adheres to the solid surfaces so that it does not slip at the plane of contact, which also is found to be true in practice.

If we apply these general ideas to a shaft bearing in which *A* is the shaft, *B* the bearing, and *O* the oil layer, the same general conclusions hold, but the geometry is different and leads to the familiar and somewhat complicated theoretical equations.

Fig. *B* shows a cross-section of such a bearing. If *A* carries a load *P* the shaft will tend to settle to the bottom and, if the shaft is not rotated, will come into contact with the bearing at *C*. But, if it rotates, oil will be dragged along with it by virtue of its viscosity, pushing the shaft to some such position as noted in the figure where *d* is the point of nearest approach of the shaft to the bearing. The position of *d* and the thickness of the oil at this point will, of course, depend on load, speed, viscosity, bearing clearance, and so on, which are included in the classical equation.

But the equations, for the sake of simplicity, assume that the oil will have a uniform viscosity and that it does not leak out at the ends. Since the viscosity depends on temperature and to some extent on pressure, and since all bearings have end leakage, obviously the "theoretical" equations are only approximately true. If they are nearly enough true to be useful – well then they are useful; otherwise they are not.

Their limitations in accuracy naturally depend on conditions. If bearings are too short, they leak more at the ends; if they are too long, the shaft bends and the thickness of oil at *d* in Fig. *B* is not uniform throughout the length. The theoretical equations generally apply most nearly for a ratio of length to diameter of somewhere between ¾ : 1 and 1 : 1.

The oil film near *d* obviously contributes more to the friction load because it is thinnest there.

This portion of the film is under a pressure which may be as high as five times the unit load on the bearing as a whole. In a bearing loaded to 200 lb. per sq. in. the oil at *d* may be under a pressure of 1000 lb. per sq. in. Also it is being heated continually by the frictional heat generated due to its viscous resistance. Therefore, the oil in the bearing is *not* at a uniform temperature nor at a uniform pressure. We

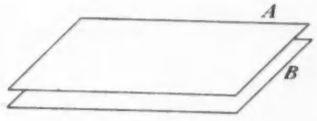

Fig. *A* (Dickinson Discussion) –Parallel Planes Separated by a Layer of Oil

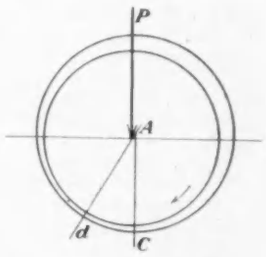

Fig. *B* (Dickinson Discussion) – Cross-Section of Shaft Bearing

know that viscosity is affected greatly by temperature and also somewhat by high pressure. Hence, the classical equations cannot be entirely valid. To make the most of them we should take for viscosity an average value giving more weight to the section near *d*. In fact, sometimes we might measure the bearing friction and calculate the oil viscosity more accurately than we can calculate the bearing friction from ordinary measurements of viscosity at constant temperature.

It seems evident that, if viscosity is affected by pressure, we should not confuse this effect with oiliness. It simply means that a *different* value of viscosity is required in the classical equations.

But oiliness may be *equally* as important as viscosity. If the load at *P* in Fig. *B* is too high or the speed too low, the journal comes into contact with the bearing. The classical equations obviously no longer hold even to the extent that they did under normal conditions. When the rubbing surfaces, either of the metals or of some adhering film, come in contact, three things are to be considered:

(1) The friction becomes, at least in part, that between solid surfaces. For this part of the friction the coefficient differs radically from the viscous friction of an oil film. It depends on the nature of the surfaces in contact and often is nearly independent of speed, instead of proportional to it. (This is the portion of the process to which the term "oiliness" has been confined by definition.)

(2) Metals tend to weld together or score each other when they come in contact, and certain sorts of adhesive film tend to prevent this scoring. With these films on the metal, loads can be carried on sliding solid surfaces, such as that of the shaft on the bearing, which otherwise would cause immediate failure from destruction of the metal surfaces. Thus we have E. P. lubricants. The term oiliness is not accepted as applicable to this factor, and we have no generally accepted word for it. It may be called load-carrying capacity.

(3) When metal or other solid surfaces rub together, there generally is wear. This wear depends on the nature of the surfaces, their resistance to abrasive particles, and various other factors. Not having the term oiliness to blanket this factor, some other word is needed, such as "wear".

Thus in the field to which the term "oiliness" has sometimes been applied, which we enter only when the shaft begins to contact the bearing, there are three separate and distinct phenomena which may have little or no relation to each other. One of them has been given the name "oiliness". These are:

(1) Oiliness: the coefficient of friction which determines how much heat will be generated and power consumed at a given speed and load.

(2) Load-carrying capacity, which determines how much load the bearing will carry at any given speed, temperature, and so on, before the metal surfaces will be destroyed by scoring, scuffing, abrasion, and so on.

(3) The rate of wear, which determines how long a bearing will last, provided it does not burn up or seize, under any particular set of running conditions.

We need good names for items (2) and (3).

Defines Fluid-Film Lubrication

– Edward R. Barnard

Standard Oil Co. (Ind.)

IN the past we have read many papers on fluid-film lubrication, usually dealing with relatively thick films in journal bearings in which moderate unit loads and rather high rubbing speeds were employed. Film rupture or failure apparently occurred under very moderate conditions as compared to the operating loads and speeds referred to in this paper. Also it usually has been assumed that boundary conditions of lubrication imply much heavier unit pressures and/or lower rubbing speeds than those used by Professor Everett, and that between conditions of boundary lubrication (affected by oily materials) and fluid-film lubrication there lies a veritable "no-man's land" called "partial-film" lubrication. This dark region has usually been considered a ghostly place, frequented by such bugaboos as "instability," "erratic behavior," "incipient seizure," and so on. It appears that Professor Everett has done much to disprove the existence of these spooks.

In some of our work on friction under high loads and low rubbing speeds, we have used bearings comparable both in size and form to that described in this paper. We also noted that, when the surfaces were fully generated, fluid films could be maintained at a load of 5000 lb. per sq. in., with rubbing speeds as low as 4.5 ft. per min. Up to this point, the coefficient of friction would decrease with increasing load. Beyond this point, the coefficient of friction would become independent of the load (Amonton's Law), indicating that true boundary conditions had been reached. But it appears significant to us that in no case did a region of instability or fluctuation appear. We were at the time thoroughly convinced that this "no-man's land" did not actually exist, and Professor Everett has helped to substantiate this belief.

In our opinion, fluid-film conditions of lubrication exist when the separation of two rubbing surfaces is three or more times the length of a molecule of the lubricant. According to Langmuir the length of a molecule of oleic acid is about 1.1×10^{-7} cm. or 4.33×10^{-8} in. Since Professor Everett's minimum film thickness is computed at 5.52×10^{-6} in., it is seen that ample space is provided for about 1250 oleic-acid molecules, placed end-to-end between surfaces. There should be plenty of room, under the conditions indicated, for quite a few petroleum molecules, even if they are considerably larger than oleic-acid particles. It would, therefore, seem that the conditions necessary for fluid-film lubrication were maintained at all times.

True oiliness, whatever that may be, we believe to be significant only under those conditions of load and rubbing speed supporting Amonton's Law, that is, to have only a surface or boundary effect. When such materials appear to affect friction under conditions where coefficient varies with load, the true influence is in all probability a viscosity effect only, regardless of external appearances. We hope Professor Everett keeps up the good work.

Derivation of Equation for Absolute Viscosity

– W. E. Lay

University of Michigan

THIS is a mechanical age. Machines must be lubricated to operate efficiently. It is seldom indeed, when a machine is designed and operates successfully, that a major lubrication problem is not solved in the process.

I fear that many of us are floundering about in the morass of apparent inconsistencies in the behavior of different lubricants and bearing materials under the various conditions of temperature, pressure, speed, and so on, at which they may operate. The engineer and the professor, or perhaps we should say physicist, are inclined to disagree, in fact, they may not understand each other because they use such different terms and look at the problems from such widely separated viewpoints. I have the feeling, however, that this general tangle will be straightened out and our difficult engineering problem will degenerate into a simple problem of physics and chemistry leavened perhaps with a bit of mathematics.

Several years ago in the discussion of lubrication before a class, I mentioned the term centipoise. Promptly I was asked: "What is it?" And, having had experience, as promptly I replied that it was due to be discussed at our next meeting. In the meantime I started looking for that definition. I began looking in books and articles on lubrication. I finally found it in a book on hydraulics.

In the first place, absolute viscosity is not a force, nor is it a ratio of pure numbers. Correctly termed, it is the coefficient of absolute viscosity whose unit is a poise. It may be explained rationally as follows:

Suppose we consider two horizontal plane surfaces, as shown in Fig. *C*, each having an area A. They are separated to the distance Y, and the space between them is filled with oil. The application of a force R to the upper surface causes it to move with a velocity V.

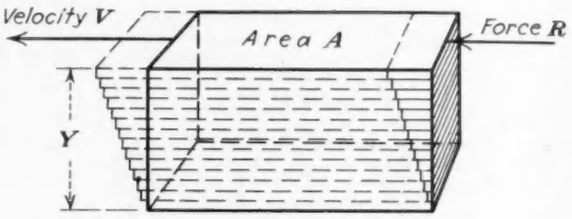

FIG. C

Fig. *C* (Lay Discussion) – Two Horizontal Plane Surfaces Separated by Oil Film

It seems reasonable to suppose that increasing the area A will require an increase in the force R thus:

$$R \propto A$$

To increase the velocity V might reasonably require an increased force R and we may expect:

$$R \propto V$$

Let us now consider the oil to be divided into a very large number of horizontal layers of equal thickness and then move the upper surface to the left, to the position shown by the dotted lines. The upper layer of oil adheres to the under surface of the upper plate and moves with it to the position shown. The bottom layer of oil is attached to the upper surface of the lower plate and does not move at all. Intermediate layers of oil move intermediate distances, each layer moving slightly with respect to the layers above or to the layer below it. When Y is greater there will be more layers, and the sliding, or shear, between any one layer and the layer above or below it will be less. Hence the shearing force required to produce this shear between any two adjacent layers will be reduced and we may reasonably expect that:

$$R \propto \frac{1}{Y}$$

Collecting the above proportionalities into one expression we have:

$$R \propto \frac{AV}{Y}$$

This proportionality may be changed to an equality by the introduction of the proper coefficient. That proportionality coefficient is "μ" the *"Coefficient of Absolute Viscosity"* and the equation then becomes:

$$R = \mu \frac{AV}{Y}$$

$$\frac{R}{A} = \mu \frac{V}{Y}$$

$$\mu = \frac{\frac{R}{A}}{\frac{V}{Y}} = \frac{\text{Unit shear stress}}{\text{Velocity gradient}}$$

If the force R is one dyne,
the area A is one square centimeter,
the velocity V is one centimeter per second,
the distance Y is one centimeter,
then the coefficient of viscosity "μ" is one poise.

This equation may be applied to the simplest case of a shaft running free without load in a bearing with thick-film lubrication as shown in Fig. *D*.

Bearing length is l cm.
Diametral clearance is c cm.
Shaft diameter is d cm.
Shaft speed is n r.p.s.

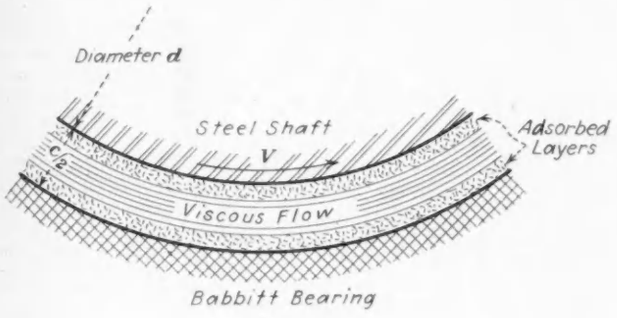

FIG. D

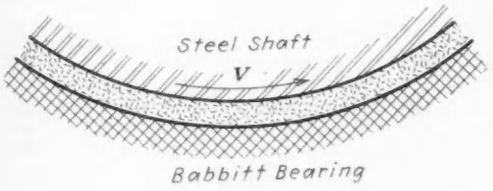

FIG. E

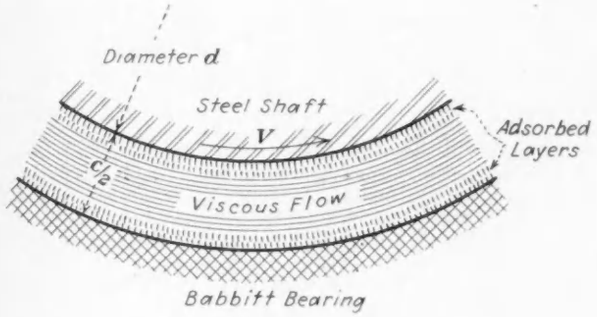

FIG. F

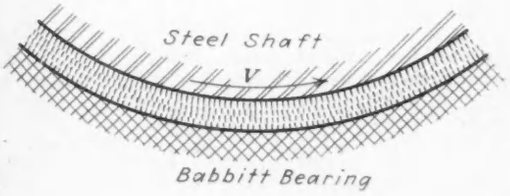

FIG. G

Fig. *D* (Lay Discussion) – Thick Film – No Addition Compound – Adsorbed Layers Not in Contact – Friction Force Determined by Viscosity of Oil

Fig. *E* (Lay Discussion) – Thin Film – No Addition Compound – Adsorbed Layers in Contact – Friction Forces Greatly Increased Due to Cohesive Forces Between Metal and Lubricant Molecules

Fig. *F* (Lay Discussion) – Thick Film with Addition Compound – Adsorbed Layers Not in Contact – Friction Force Determined by Viscosity of Oil – Molecules of Adsorbed Layers Regimented in Regular Order Perpendicular to Metal Surface

Fig. G (Lay Discussion) – Thin Film with Addition Compound – Adsorbed Layers in Contact – Easy Slippage along Cleavage Lines between Layers of Regimented Molecules – Friction Force Not Greatly Increased

$$\text{Area } A = \pi d l$$
$$\text{Velocity } V = \pi d n$$
$$\text{Distance } Y = \frac{c}{2}$$
$$\text{The friction force } R = \mu \frac{AV}{Y} = \frac{2\pi^2 d^2 l n \mu}{c} \text{ dynes}$$
$$\text{The friction torque} = R \frac{d}{2} = \frac{\pi^2 d^3 l n \mu}{c} \text{ dyne-cm.}$$

When the load is applied to the shaft, then the clearance is no longer uniform around the bearing and the problem becomes more complicated and will not be considered here. The coefficient of viscosity is thus a physical property of the liquid. As I understand it, it is a measure of the forces by which the molecules of the liquid attract each other. That is, it is a measure of the forces of cohesion, which are the attracting forces between like molecules. The molecules in the boundary layer attached to the metallic surfaces of shaft and bearing are affected by another type of force, the force of adhesion, the attracting forces between unlike molecules. The adsorbed boundary layers are very thin, perhaps a millionth of an inch, being composed of only a few layers of molecules held to the surface by a force greatly exceeding the force of cohesion. If a greatly increased load is applied, the thickness of the oil film is reduced to the point where these adsorbed layers begin to engage each other as shown in Fig. *E*. Due to much greater adhesive forces and the interlocking of the molecules, these molecules resist any dislodging action and the resistance to turning the shaft becomes much greater.

The addition of certain compounds to oils seems to increase the load capacity of the bearing without increasing the friction. By rapidly revolving a cylinder on whose outer surface this oil has been placed, we find that these oils will cling to the surface resisting much greater centrifugal forces indicating greatly increased adhesive forces between the molecules of the oil and the molecules of the metal.

An interesting and rather reasonable theory has been advanced to show why the friction has not been increased by the addition compounds even though the adhesive force and bearing load capacity is increased greatly. It is known that the paraffin molecule is long and narrow somewhat like a saw log. Examination of this thin film by X-ray indicates that the molecules lie in regular layers all standing on end with respect to the bearing surface and are said to be regimented as shown in Fig. *F*. Since the lengths of the molecules are the same, their ends reach the same height above the surface and present a line of cleavage. When the adsorbed layers are in contact, as in Fig. *G*, the forces necessary to slide one layer of molecules over the other along this cleavage line may be small compared to that required to dislodge the molecules from the heterogeneous log-pile jam occurring when the addition compounds are not present. Thus, in thick-film lubrication, the viscosity or the cohesive force is the important factor whereas, in thin-film lubrication, the adhesive force is most important.

At any rate there is indication that much may be learned from a careful examination of the molecular structure of these very thin, adsorbed boundary layers. This is a job for the physicist.

Similar Tests on E.P. Lubricants Reported

– F. A. Faville

Faville-LeVally Corp.

THE discussion of whether or not test pieces were in full-fluid or boundary film is the very essence of the testing problem. We know that bearings, under forced lubrication and in true fluid film operate with almost no wear at all. Professor Everett brings out that all of his work was carried out with the same set of matched bearings and with no apparent wear although unit loadings of 30,000 lb. per sq. in. were used.

In experimenting at high unit loadings and varying speeds, we have found that, when boundary film is established, the time rate of wear, and the heat at the point of contact, are accelerated many times the rates prevailing in full-fluid film. In the range between boundary and full-fluid film, it is possible to produce accelerated wear. Some partial metal-to-metal contact, which continually penetrates the oil film and produces wear, appears to exist in this range.

Some addition agents appear to disturb the relationship existing for straight mineral oils, and maintain what is apparently full-fluid film, even though the rate of wear is accelerated greatly. There is likelihood of considerable variation when working in this range, depending upon the percentage of metal-to-metal contact. The attempted acceleration of wear rate when testing in this range may readily produce results out of all proportion to actual results in service but, by proper care and control of testing, we believe that this type of test can be correlated to service results with much better success than seizure-type tests on any laboratory machine, and the reproducibility of results is far better.

The work that Professor Everett has presented is most interesting. In an extended series of tests made by Max Jester and the writer, a year ago, we endeavored to determine this ΔT, by using a fine-wire thermocouple in the base of the V-blocks in studying the temperature effects on extreme-pressure lubricants. With oils under higher unit pressures than Professor Everett has employed, readings of ΔT of 150 deg. fahr. difference in temperature were obtained with apparent fluid film still maintaining. At the moment of pop-up or incipient seizure, this difference increased almost immediately to 300-400 deg. fahr. Wear tests in boundary film showed a total temperature at the base of the V-blocks of as high as 700 deg. fahr. with a lubricant containing free sulphur. The rate of wear is accelerated over 100 times when boundary film is established. Some products stay in boundary film once it is established, and some show a persistent tendency to return to apparent fluid film, with return to the temperature, friction, and wear prevailing before boundary film was established. Boundary films can be established at machine loads as low as 700 lb. jaw load with abrasive-type lubricants on wear type tests. Our studies indicated the coolant effect of the lubricant in preventing development of welding temperatures to be a predominant characteristic in the behavior of the lubricant.

Compares Various Concepts of Oiliness

– James I. Clower

Associate Professor of Machine Design, Virginia Polytechnic Institute

I BELIEVE that Prof. Albert A. Kingsbury was among the first to call attention to that elusive, intangible something commonly called oiliness. In his paper: "A New Oil Testing Machine and Some of Its Results,"[a] presented before the American Society of Mechanical Engineers in 1903, he made the following pertinent statements:

"It has become well recognized by experimenters that there are two important properties of lubricating oils and greases on which their value depends. Viscosity is the property by virtue of which the lubricants form comparatively thick films between rubbing surfaces, permitting perfect lubrication. A condition essential to the formation of such films, as shown by Reynolds, is that the rubbing surfaces should have a very slight inclination to each other in the direction of their relative motion. When the loads are very great or the surfaces irregular, or when the conditions are otherwise such as to make the necessary inclination impossible, it is well known that the action of the lubricant is imperfect. Under any of these circumstances the effect of the lubricant in reducing friction depends mainly upon the "body" or "oiliness." The nature of this property, or combination of properties, is not well understood, but it appears probable that it is an *intensified viscosity* in that part of the fluid within the region of the attraction of the surface molecules of the metal."

In addition to intensified viscosity as a possible explanation of oiliness many other theories have been offered by various investigators from time to time. Some of these may be described briefly as plasticity, secondary bulk effect, absorption, adhesion, and stratification. Many subdivisions and combinations of these have also been proposed in an effort to explain the phenomenon of oiliness.

At least until very recently Herschel's definition of oiliness has been accepted widely as representing the practical engineering point of view of oiliness; yet his definition does not entirely dissociate oiliness from viscosity in a true physical sense. Mr. Herschel defined oiliness in his paper, "Viscosity and Friction,"[b] as, "the property that causes a difference in the friction when two lubricants of the same viscosity at the temperature of the oil film are used under identical conditions."

Since Mr. Herschel did not specify that the lubricants must have the same viscosity at the pressure in the film, lubricants are eligible for comparison under his definition if they have equal viscosity at the temperature of the film when the viscosity is determined at atmospheric pressure in the usual way. Therefore, the differences in friction of two oils under load that are matched as to viscosity at the operating temperature and atmospheric pressure but have different pressure viscosity coefficients could be attributed to oiliness. In fact, however, the differences in friction would doubtless be mainly caused by the differences in viscosity produced by pressure.

The present-day concept of oiliness, I believe, is that it is a property or properties independent of viscosity. This concept is embodied in the SAE Crankcase Oil Oiliness Research Committee's definition, which states that "Oiliness is a term signifying differences in friction greater than can be accounted for on the basis of viscosity when comparing different lubricants under identical test conditions." This definition, I believe, recognizes that viscosity and oiliness are two distinct characteristics of a lubricant. If and when we discover the true nature of oiliness it may be found, however, that viscosity and oiliness are related.

In the light of the Crankcase Oil Oiliness Research Committee's definition of oiliness, Professor Everett's results are in no way related to oiliness unless it can be shown that the differences in ΔT and, hence, friction are greater than can be accounted for on the basis of viscosity alone.

It has long been known that high pressure produces measurable increases in the viscosity of liquids. Among the first to do experimental work along this line was Prof. C. Bares, of Brown University, who in 1893 observed that the viscosity of marine glue increased with pressure. In 1901, C. Hauser of Germany conducted extensive pressure-viscosity tests with water, and in 1916, M. D. Hersey of this country, using the Flower's ball-and-tube type of viscosimeter performed similar tests, using various lubricating oils. Further experiments have been conducted by Hyde of England, Kleinschmidt, Hersey,[c] and Shore of this country, Kiesskalt of Germany and Suge of Japan.

Professor Everett states in his paper that "the high-pressure data show so clearly the great differences that exist in different oils that it seems obvious that data of this sort are as essential as the temperature data when considering the performance on an oil bearing. Without the P.Z.T. charts or their equivalent we cannot explain or predict oil-film performance in bearings." I doubt very much if P.Z.T. charts are necessary when considering the performance of an oil in that class of machine elements which we are accustomed to call bearings. Few if any automobile bearings are designed for maximum pressures in excess of 1500, or at the most 2000 lb. per sq. in. of projected area, although the maximum pressure within the film itself is, of course, much greater than any of these values. Nevertheless, the pressure coefficient of viscosity is, in my opinion, relatively unimportant compared to that of temperature for pressures of these magnitudes.

If, on the other hand, Professor Everett has in mind the performance of a lubricant for gear teeth, cutting tools, metal drawing dies, and so on, undoubtedly the pressure coefficient of viscosity is of importance and P.Z.T. charts are essential. In fact, if the pressure is extremely high, the pressure coefficient of viscosity may be of greater importance than that of temperature. Undoubtedly, were it not for the increased viscosity due to pressure, we could not successfully lubricate such parts with the relatively light lubricants that are used.

With reference to Figs. 12 and 13 of Professor Everett's paper it seems to me that, although an oil particle is at the instant it enters the bearing subject to a pressure of 20,000 lb. per sq. in., it is at the same time subject to the temperature of the bearing. Thus, although the pressure would tend to cause an increase in viscosity, the temperature would tend to cause a decrease and, therefore, the net change in viscosity would be the difference between these two conflicting factors. For example, consider the western oil in Fig. 13, its viscosity is apparently increased from 40 to 1830 centipoises due to pressure but, at the same time, it probably is reduced by temperature to 190 centipoises. This condition is, of course, assuming that viscosity reacts instantaneously to both pressure and temperature.

We have, however, no assurance that the constantly varying pressure and temperature conditions within a bearing have the same influence on the properties of a lubricant as has a uniform hydrostatic pressure such as is used in determining the viscosity-pressure relationships and viscosity-temperatures in the laboratory. In a bearing a particle of a lubricant is subject to varying intensity of both the pressure and temperature as it moves through the bearing. A very vivid picture of these changing conditions has been given us by Professor Everett when he permitted us to peek behind the shades into the "private life" of an oil particle as it staggered along the tortuous path through a bearing. We know that, although every liquid has a definite viscosity value corresponding to any particular temperature; yet, when the temperature is changed rapidly, the corresponding change in viscosity is not necessarily instantaneous and sometimes takes a measurable and appreciable time to settle down to a constant value. This peculiar lagging effect is known as hysteresis and, although not commonly met with to a noticeable extent, is occasionally exhibited by oils and other liquids whose molecules are believed to be highly associated.[d] Perhaps a similar phenomenon occurs in connection with the application of pressure and, if so, the combined temperature and pressure hysteresis effect may be quite pronounced.

In conclusion, I cannot help but wonder what trends the oils used by Professor Everett would have exhibited had they been subjected to a continuous oxidation process. Certainly the oil one puts into a crankcase of an engine does not retain its initial characteristics for any appreciable length of time. Its character is changing constantly and, in some respects, for the better. Any refiner would have considerable difficulty identifying his own product after it had rendered a few hundred miles of service in the crankcase of an engine. So, it does not seem to me that the most important consideration is so much the initial character of the oil, but rather the average character considered over its period of service.

In conclusion, I should like to ask Professor Everett just how he arrived at the correction temperatures of 4, 5, and 11 deg. fahr.? It seems to me, since the bearing was undoubtedly cooled by conduction, convection, and radiation that, unless some elaborate system of temperature measurement were devised, an estimate would be little more than a guess. Furthermore, I do not understand what are to be considered the correct viscosities given in Table 2 at 14.2 lb. per sq. in. pressure.

[a] See A.S.M.E. Transactions, Vol. 24, 1903; pp. 143-160: "A New Oil Testing Machine and Some of Its Results," by A. A. Kingsbury.
[b] See S.A.E. Transactions, Vol. 17, Part I, 1922, pp. 282-320; "Viscosity and Friction," by W H. Herschel.
[c] See "Theory of Lubrication," by M. D. Hersey, John Wiley and Sons, Inc., 1936.
[d] "Lubrication and Lubricants," by Archbutt and Deeley, Griffin, London, 1927.

Study of Additional Characteristic Recommended

– G. H. B. Davis

Esso Laboratories, Standard Oil Development Co.

PROFESSOR EVERETT is to be congratulated for pointing out so clearly that there are other factors besides the viscosity-temperature relationship of oils that have a profound influence upon friction. Also, he has clearly demonstrated the fallacy of classing all these other properties under the term "oiliness."

Work of this type is very useful in helping the engineer explain the results he obtains with a particular machine and also in aiding the petroleum engineer in developing oils that will give the best lubrication with the least friction under operating conditions.

I do not interpret this paper as indicating that oiliness is not still a problem but rather that the field covered by oiliness has been narrowed. One other characteristic, which may not be as important as the viscosity-temperature relationship or the viscosity-pressure relationship still requires further study, namely the effect of rate of shear upon viscosity at different pressures. If data are available for these three properties of oils then their performance in an operating machine can be more closely understood.

Discusses Possible Effects of Shear Rate

– G. L. Neely

Standard Oil Co. of Calif.

PROFESSOR EVERETT'S paper presents an interesting interpretation of the viscosity data obtained at high pressures by Dr. Dow. It is obvious that these high-pressure viscosity data can be obtained only in carefully designed and well operated equipment, and Dr. Dow deserves special credit for this work.

One of our greatest needs in the study of lubrication is information on the change of viscosity with pressure and methods of applying these data. Dr. Dow has supplied some of the necessary data, and Professor Everett has indicated a method of applying them. In order to understand fully the phenomena involved and be able to appreciate their practical aspects the following questions seem pertinent:

(1) How do oils of the same family but of different viscosity react to pressure changes?

(2) How do fixed oils react?

(3) How does the rate of pressure application where the pressure varies such as in an automotive bearing affect the viscosity?

We do not have the equipment available in our laboratories for studying viscosities under high pressure so I can offer no further comment on this type of work, except that it appears to throw light on some of the fundamental behaviorisms of lubricating oils and undoubtedly will be of value in showing why some experimental data are seemingly at variance with the laws and theories generally used at least up to the present time. If this new variable pressure tends to clarify many of these problems perhaps we will have a pressure index and be talking about "P. I." as well as V. I.

One of the factors that may affect viscosities under operating conditions which was not mentioned by Professor Everett is shear rate. In 1931 Averson showed and discussed data relative to the marked effects of shear rate on the viscosities of greases whereas, in the next year, Kyropoulos, who worked with a tapered-plug, concentric-cylinder viscometer, showed that mineral oils also may be so affected. We also have obtained some data on the subject of shear rate and, although we are not prepared to draw conclusions, we feel that the matter of shear should not be overlooked in such studies as those presented by Professor Everett. It is significant that Kyropoulos' shear rate was in the order of 50,000 reciprocal sec. whereas, in Professor Everett's machine, the shear rate would be 600,000 or more reciprocal sec. The combined effects of high pressure and high shear rate on the viscosities of lubricating oils might be particularly pertinent.

It is noted that the oils chosen for this work were not necessarily representative. This condition is particularly noticeable for the case of the western oil which had a viscosity index of —20, whereas western oils of viscosity indexes of greater than +20 are now being marketed commonly. As the type and degree of treatment materially influence the lubricating value as well as most other properties, it would be very interesting to know if Professor Everett had ever tested a finished western oil of the type now normally sold.

In regard to Professor Everett's data, it is assumed that there is no metallic contact between the bearing and the shaft since the distances between these members has been calculated. It would be interesting to know, however, if any attempt has been made to determine experimentally that these surfaces are actually separated. We have used direct-current circuits with suitable relays to indicate metallic contact in several of our demonstrating units and have had remarkably good success with such devices. Another question that might be asked is the degree of reproducibility obtained between different runs on the same oil. It would seem that for conditions of fluid-film lubrication with no wear taking place that checks should be excellent. This brings up another point: Do the test blocks or the rotating cylinders wear to a measurable extent? In view of the present-day interest in compounded oils, do addition agents that increase oiliness in other types of measuring equipment without influencing the Saybolt tube viscosities have any effect in this test?

Also, according to hydrodynamic theory the type of metals used has no influence on the results. It would, therefore, seem pertinent to ask if the effect of metal combinations such as hardened steel on hardened steel have been investigated by Professor Everett. In this case scoring would be the criterion of metallic contact.

An interesting interpretation of Professor Everett's data has been made showing the viscosities of the oils at the initial and final temperatures both for the three different type oils and for a series of eastern oils showing the same temperature rises. These data are as follows:

		Viscosity, centipoises			
Oil	Temperature Difference, deg. fahr.	Viscosity at 130 deg. fahr. at 14.2 lb. per sq. in.	Viscosity at 130 deg. fahr. + Temperature Difference at 14.2 lb. per sq. in.	Viscosity at 130 deg. fahr. at 20,000 lb. per sq. in.	Viscosity at 130 deg. fahr. + Temperature Difference at 20,000 lb. per sq. in.
Western	53	43	12	1,830	198
Eastern	53	240	45	3,250	356
Mid-continent	38	41	17	740	210
Eastern	38	82	34	1,100	303
Eastern	27	39.5	23	525	223

It is noted from the foregoing that eastern oils causing the same temperature rises are higher in viscosity. Perhaps shear rate as mentioned previously is the cause of the apparent discrepancy between theory and data, or perhaps it is only the false assumption that the pressure is constant over the bearing area that causes this discrepancy.

The practical aspects of the effects of the very high pressures appear to relate primarily to gearing. In this regard our information is apparently in line with the viscosities shown since it generally has been our experience that uncompounded western oils are superior to uncompounded eastern oils for steel-on-steel gearing. This condition has been found to be true both for steel-on-steel spur gears and for steel-on-steel worm gears and, as the pressures involved were apparently of the order described in the paper, the increased viscosities offer a reasonable explanation.

A practical interpretation of Professor Everett's findings as presented would be that, to obtain the same bearing protection or minimum oil-film thickness, a lighter grade of low-viscosity-index oil should be used. This method would be of material advantage in most practical applications since the lightly loaded parts would have a low-viscosity oil as is desirable, whereas the relatively greater viscosity increase in viscosity of the western oil through pressure would provide the necessary safety factor for the heavily loaded sections. The pressure-viscosity characteristics which the author finds so favorable to low-viscosity-index oils should more than offset the advantages of high viscosity index for many types of lubricating practice.

Suggests Improvements in Test Machine

– A. W. Burwell

Alox Corp.

I AM somewhat inclined to believe that Professor Everett's method of test throws the whole matter over into the realm of viscosity rather than keeping definitely within the conditions of test which can and we believe do show the effect of oiliness as distinguished from viscosity. In the first place we see no statement in the paper regarding the definite fitting that is parallelism of the surfaces opposed to each other. The size of the surface employed upon the bronze or brass bearing block is, in our estimation, too small to permit of proper accurate fitting. The pressures employed we have found to be impossible in those machines which, in our opinion, are capable of contributing accurate data to the question of oiliness.

There is one other feature which has not been touched upon by anyone making contributions to this particular subject and that is a definite statement that the film, or films, upon the metal surfaces, more particularly in so-called highly oily oils, are definitely distinct in composition, viscosity, and other constants from the oil as a whole. This feature might, in part, be an explanation of the apparent differences found and stated in Professor Everett's paper. We doubt seriously that real evaluations upon surfaces that are too small, or in machines where

velocities are too high, can be made and the finer distinguishing points concerning the differences between the protective power of viscosity as compared with oiliness, can be brought out. We wish to note, however, that the data as presented by Professor Everett have definitely placed the majority of oils derived from the various crudes obtained in this country in the same order of preference that we have found in our work with the Lubarometer. We distinctly, therefore, do not even deprecate the use of such a machine, but we definitely believe that it does not go far enough nor is it capable of giving the wide variety of results obtainable by the use of more carefully controlled speeds, temperature, and other factors so necessary in the requirement of accurate data.

Proposes Method for Comparing Oiliness of Oils

– O. C. Bridgeman

National Bureau of Standards

AT a meeting of the SAE Crankcase-Oil Oiliness Research Committee held in Detroit at the time of the 1937 Annual Meeting, the Committee adopted a definition for oiliness which was stated as follows: "Oiliness is a term signifying differences in friction greater than can be accounted for on the basis of viscosity when comparing different liquids under identical test conditions."

It follows from this definition that both the temperature and pressure coefficients of viscosity must be taken into consideration when comparing frictional data obtained on different oils, and I am pleased to note the emphasis which Professor Everett has placed on the necessity for considering the effects of temperature and pressure on viscosity when evaluating oiliness. Many investigations have recognized the effect of temperature on viscosity and, in most cases, have made allowance for this effect by comparing oils of the same viscosity at the temperature existing in the oil film between two surfaces moving relative to one another. Too little emphasis, however, has been placed on the effect of pressure on viscosity and I believe that Professor Everett is doing a great service in pointing out that many values hitherto obtained for oiliness are actually only apparent values and are in reality partly or wholly attributable to different pressure coefficients of viscosity. It is true in many cases that the effect of pressure on viscosity is not of primary importance from the standpoint of numerous service applications, either because the pressures are not very high or because the addition of a small amount of compounding agent may not produce any marked change in the pressure coefficient of viscosity. On the other hand, particularly when considering gear lubrication under heavy load, differences in pressure coefficients of viscosity may produce marked differences between the friction values obtained with various oils.

Oiliness is of major interest at present in connection with cylinder-wall and piston-ring lubrication. Under these conditions, it is regretted that much of the experimental data presented by Professor Everett is

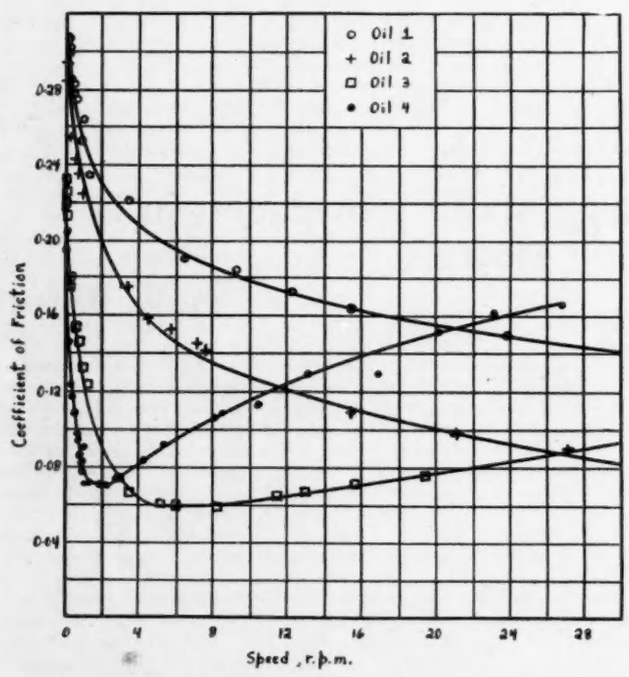

Fig. *H* (Bridgeman Discussion) – Coefficient of Friction Vs. Speed at Constant Load for Four Oils of Different Viscosities – Thin-Film Lubrication

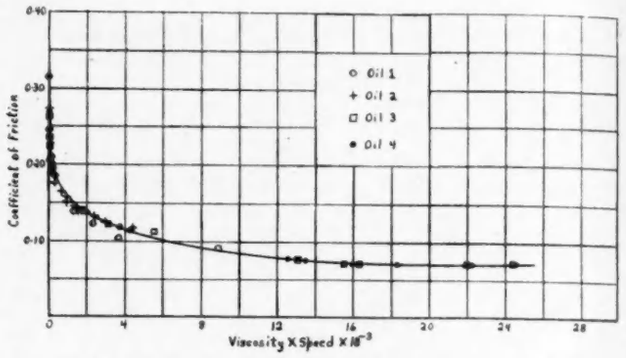

Fig. *I* (Bridgeman Discussion) – Coefficient of Friction Vs. Product of Speed and Viscosity for Four Oils of Different Initial Viscosities

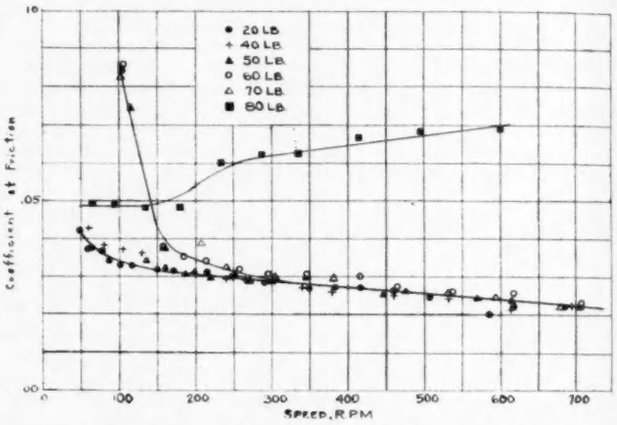

Fig. *J* (Bridgeman Discussion) – Coefficient of Friction Vs. Speed for One Oil Operated at Various Loads

applicable to higher pressures than those of interest in connection with cylinder wall lubrication or even bearing lubrication. Thus, under the conditions existing in automobile or aviation engines, the pressures are sufficiently low so that differences in the pressure coefficients of viscosity between oils of widely different source are small as compared with the differences shown by him at high pressures. Accordingly, although proper consideration should be given to the change of viscosity with pressure, in most cases of cylinder-wall lubrication, differences found for the oils being used at present, particularly in aviation service, are so small as to be relatively unimportant.

There is one point in connection with the data on temperature rises to which I would like to refer in somewhat more detail. It was shown that the oil with the lowest viscosity at the temperature existing in the oil film showed the highest temperature rise under the conditions of test, and this phenomenon was accounted for on the basis of change of viscosity with pressure. There are conditions, however, where this same effect has been found at pressures so low that the differences could not be accounted for on the basis of the pressure coefficient of viscosity. Let us assume that an increase in temperature rise of the oil entering and leaving the bearing can be considered as indicative of increased friction, or of increased coefficient of friction at constant load. It follows from our hydrodynamical theory that the coefficient of friction increases with increasing viscosity in the region of thick-film lubrication. On the other hand, in the region of thin-film lubrication, the coefficient of friction decreases with increasing viscosity, at least at moderate loads. This point is illustrated in Fig. *H*, which shows coefficient of friction data at constant load for 4 oils having the following viscosities:

Oil	Viscosity, centipoises
1	41.7
2	233
3	3120
4	9340

To the left of the minimum point, at any given speed, the oil with the higher viscosity has the lower coefficient of friction. Considering the data now in terms of temperature rises, the oil with the lowest viscosity would show the highest temperature rise and the oil with the highest viscosity would show the lowest temperature rise. Further, the load was so small that all of the differences cannot be explained on the basis of differences in the effect of pressure on viscosity. Although the data presented in Fig. *H* were obtained under one set of conditions, values obtained under numerous other conditions with other testing machines, metal combinations, speeds, loads, and oils indicate the same conclusion that in the thin-film region the coefficient of friction and hence the

temperature rise in the oil film increases with decreasing viscosity. This conclusion is being restricted to data obtained under conditions where there is a wedging effect in the oil film, which is the condition of interest in connection with engine lubrication, and where loads are not greatly in excess of those encountered in engine bearings or on cylinder walls. Under conditions where there is no wedging effect or where loads are very high, the situation may be different.

The definition adopted for oiliness by the SAE Crankcase Oil Oiliness Research Committee states that temperature and pressure coefficients of viscosity must be taken into consideration when evaluating oiliness, but no mechanism is proposed for taking these effects into consideration. Although insufficient data are available to establish definitely any such mechanism, one method can be proposed which has been very effective in a number of cases. In making this proposal, no consideration is given to friction data obtained under conditions where there is no wedging effect in the oil film, and it is difficult to see why we should be interested in any other set of conditions when considering engine lubrication. Going back to the Bureau data presented in Fig. *H*, if the viscosity of the oil at the temperature in the oil film is evaluated for each observation and a plot is made of coefficient of friction against the product of the speed and the viscosity (at constant load), a single curve is obtained, as illustrated in Fig. *I*. In other words, the effect of differences in viscosity has been eliminated and it is observed that no appreciable differences in oiliness exist between these 4 oils under the test conditions. A somewhat more striking example is shown in Fig. *J* for one oil operated at various loads and speeds, but where marked differences in temperature exist in the oil film. These data were obtained about 5 years ago with a machine almost identical with that used by Professor Everett, as an outgrowth of the Bureau work on extreme-pressure lubricants. Again evaluating the viscosity in the oil film for each observation and using the product of the speed and viscosity, a single coefficient of friction curve is obtained, as is shown in Fig. *K*.

Although all of the data obtained at the National Bureau of Standards have been under conditions of moderate load and, hence, no corrections were made for the change of viscosity with pressure, there is no reason to believe that evaluation of both temperature and pressure effects should not be as successful as evaluation of temperature effects alone. In this case, in evaluating oiliness, the coefficient of friction would be plotted against the product of the speed and the viscosity of the oil at the pressure and temperature existing in the oil film. By this method, it is possible to compare the oiliness of two oils of different viscosities at the standard test temperature, of different temperature coefficients of viscosity, and of different pressure coefficients of viscosity. An example of such a comparison is given in Fig. *L* for mineral oils, lard oil and castor oil, where correction was made for the temperature coefficient of viscosity but not for the pressure coefficient, as the pressures employed were low.

Whether the method proposed for eliminating the temperature and pressure effects on viscosity is generally valid must await further investigation. However, for the time being, it is believed that it presents the most rational means for comparing the oiliness of oils. It will be observed that this method of presenting friction data may result in a series of curves for different loads. This series appears necessary at present, since the relation between coefficient of friction and load in the thin-film region is much more complicated than the analogous relation to speed. Further work may however produce considerable simplification in this regard.

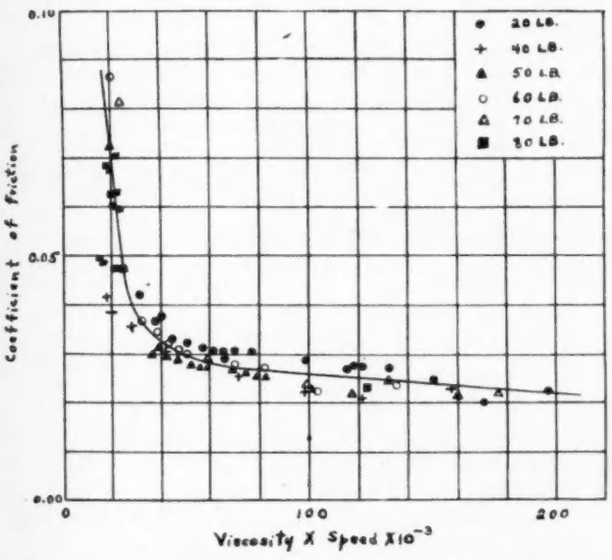

Fig. *K* (Bridgeman Discussion) – Coefficient of Friction Vs. Product of Speed and Viscosity for One Oil Operated at Various Loads

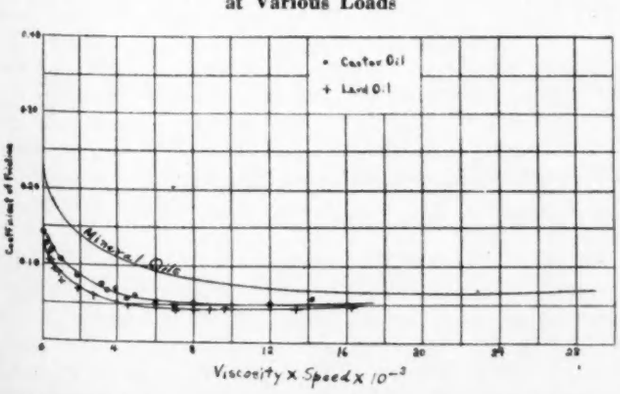

Fig. *L* (Bridgeman Discussion) – Comparison of Oiliness of Mineral Oils, Castor Oil, and Lard Oil

Confirms Author's Figures on Film Thickness

– G. B. Karelitz

Professor of Mechanical Engineering, Columbia University

ONE might consider the direct application of Professor Everett's results to practical design of bearings as used at present. In our design of bearings, or rather in the analysis of designs made by engineers, it is customary to estimate the film thickness and friction on the basis of an estimated uniform viscosity of the oil in the film. The tables presented by Professor Everett show that, for pressures up to 1000 lb. per sq. in., the increase in viscosity is not large. True, it is appreciable, but not sufficient to influence the computations enough to make them invalid for practical application.

However, substantial progress in the complicated question of lubrication cannot be obtained unless the action of the lubricant under extreme conditions is well understood. From this viewpoint, the information presented by Professor Everett is most valuable. The fact that oils from different fields behave differently under the same conditions may be significant. Fig. 4 of his paper shows that, under the same pressure, the western oils show a considerably higher temperature rise than eastern oils. It is not clear from casual observation and preliminary computations that the differences in viscosity, as shown by Figs. 9 to 11, can account for the differences in friction, if the friction is proportional to the temperature rise.

The writer recently made some experiments to measure the minimum film thickness when metal-to-metal contact begins. These experiments were described in the April, 1937, issue of the Transactions of the American Society of Mechanical Engineers. It appears that for brass-on-steel this film thickness is of the order of 0.00003 in., which is very close to the figures estimated by the author. It may happen that further analysis will show that the rather pronounced upswing of the ΔT curves at high pressures was caused by a transition from pure viscous friction to a semi-fluid state of lubrication. There is no reason to expect that such a transition would cause a break in the smoothness on ΔT curves. On the other hand, it might be that some inherent qualities of the different lubricants manifest themselves on high pressures, such as were used by the author in his experiments. The very scant evidence obtained by the writer in the above mentioned minimum film-thickness experiments was indicative of a marked difference in the behavior of various oils in the region of transition from semi-fluid to fluid lubrication.

It would be very valuable and important for the art if the author could find an opportunity and facilities for continuing his investigation, and make a thorough analysis of the data collected by him.

Makes Suggestions To Bring More Accurate Results

– K. J. DeJuhasz

Associate Professor of Engineering Research, The Pennsylvania State College

WITH the trend towards increasing bearing pressures which characterizes the present development of engines and machines, the property of lubricants denoted as "oiliness" assumes an increasing importance. Yet it has been admittedly an unsatisfactory engineering concept, qualitative and comparative rather than quantitative, and unsuited for expression in fundamental units of measurement. This concept was introduced to save the theory from a breakdown. From the definition of viscosity, bearings *should have* equal friction when lubricated with lubricants of the same viscosity irrespective of the other properties of the lubricants used. When experiments disproved this expectation it was a

severe blow to the theory and the word oiliness was invented to cover up rather than dispel our ignorance of the discrepancy.

The author's paper represents a valuable contribution towards clarifying the phenomenon of oiliness, and replacing that hazy and unsatisfactory concept with the rigorously defined term of viscosity. It is to be hoped that this auspicious beginning will be followed by a more intensive and extensive investigation for which there is a real need and for which the facilities at Penn State are exceptionally well suited. Evidently, more elaborate equipment will be needed for a more accurate investigation and, in this connection, the writer offers the following suggestions:

(1) A larger lubrication machine with a large test block would yield more accurate results, because the radiation losses and errors would be smaller, and it would make possible the use of a number of thermocouples with a closer approximation to the true temperature rise.

(2) The method of measuring the frictional resistance by means of measuring the temperature difference is open to several errors. If a machine would be constructed for measuring the torque directly, then the accuracy would be greatly increased thereby.

In spite of the evident difficulties in the way of conducting an extensive investigation with the necessary high accuracy, in the writer's opinion, it is highly desirable to carry out such an investigation, and determine whether the variation of viscosity with pressure accounts totally or partially for the phenomenon of "oiliness" and what, if any, other influences play a part in it.

Influence of Solvent-Treating

– M. Fairlie
Sinclair Refining Co.

THERE are a few points in connection with the data concerning which we would be interested to hear further explanation. Professor Everett's data could be interpreted to indicate that, with increasing paraffinicity, we get a reduction in ΔT and in pressure/viscosity relationship. The question remains, however, whether the differences are due entirely to pressure/viscosity coefficient or partly to oiliness. We wonder whether he has any data comparing solvent- and non-solvent-treated oils from the same crude source. Such data might give useful indications as to whether any differences in ΔT would be a function of pressure/viscosity coefficient or, possibly, of natural oiliness. On solvent treating, does the pressure/viscosity coefficient become further reduced with resulting increase in paraffinicity, and what is the effect on ΔT?

We are interested to know if Professor Everett feels that the differences in molecular weights and specific heats of the lubricants may have had an appreciable influence on the results obtained.

Effect of Polar Materials Discussed

– F. L. Garton and H. L. Kemmerer
Shell Petroleum Corp.

PROFESSOR EVERETT'S paper draws attention to a phase of the oiliness problem which, so far, has received little attention. The popular theory assumes that oiliness is due to preferential adsorption on metal surfaces in bearings, and so on. Reference to Figs. 9, 10, and 11 of Professor Everett's paper shows that the effect of pressure on viscosity only becomes considerable at rather high pressures, above those normally encountered in the lubrication of internal-combustion engines. Furthermore, the effect of viscosity becomes much smaller at higher temperatures, such as are found in this service. This is probably one of the reasons why it is so difficult to show differences in friction horsepower in actual engines when using oils matched in viscosity, but made from different crudes. In this connection it would be very interesting to know if Professor Everett has any data on the pressure-temperature-viscosity relationships of non-polar oils and blends of the latter with oiliness dopes, such as stearic acid. Furthermore, does he have any information as to the variation of ΔT with oil inlet temperature in the Bradford machine over the temperature range encountered in practice?

In this connection, Tausz and Szekely[e] have shown by capacitance methods that films of non-polar compounds of 12,000 Ångstrom units ruptured under stress whereas, under the same conditions, polar films as thin as 1100 Ångstrom units did not rupture.

In actual engines, the oil circulated to cylinder walls, bearings, and so on, contains a certain amount of diluent from the fuel used and also oxidation products, sludge, and so on. It can be shown, on the Herschel machine, for example, that the coefficient of friction is reduced appreciably by the presence of small quantities of oxidation products such as are formed even with the most stable oils. How far can this condition be explained by differences in pressure-viscosity relationships?

The figures for minimum film thickness given in the paper fall within the range given by Hardy[f] as that through which the forces of attraction between solid boundary surfaces are operative. At the same time, they fall within the limits of film thickness in which, according to Kingsbury[g], the laws of viscous lubrication apply. They are, therefore, in the range in which viscous and boundary lubrication overlap.

It is possible that oiliness effects may be due either to the polar materials or to the pressure-viscosity effect described, depending on conditions.

[e] See *Zeitschrift für Angewandte Chemie*, Vol. 43, 1930, p. 570; also *Erdöl und Teer*, Vol. 9, 1933, p. 331.
[f] See *Kolloid Zeitschrift*, Vol. 46, 1918, p. 268.
[g] See A.S.M.E. Transactions, Vol. 24, 1903, pp. 143-160: "A New Oil Testing Machine and Some of Its Results," by A. A. Kingsbury.

Radio Shielding on Air Transports

(*Continued from page* 530)

point of contact during vibration provides a discharge point resulting in another form of interference. The treating or painting of metal parts before assembly frequently defeats the attempt to hold them at a common electrical level. Care should be used in the application of paint or dope to parts and locations where bonding is intended. Clamps holding conduits in place must be made secure, particularly in obscure locations.

In present-day airplanes the use of electrical-motor-driven accessories is quite general and seems to be increasing. This condition has resulted in the annoying and undesirable practice of selecting and installing such devices without regard to the troubles that they may cause in the way of interference. On devices of this kind more could be accomplished by the intelligent application of filters to eliminate interference at the source than can be gained by the addition of any reasonable amount of shielding. Once the undesired disturbance has been created it is extremely difficult to confine, and it frequently can be found imposed on electrical circuits throughout the ship. The introduction of alternating-current power supplies on larger planes now contemplated may aid somewhat, but the matter should be given careful consideration.

Aside from the need for adequate shielding and bonding a great deal could be said about the construction of radio as well as of electrical equipment. In many cases performance is limited and incorrectly charged against "interference" when the fault actually lies in the unit itself or in the method of installation. Bad brushes in motors or generators, broken or loose connections, worn insulation on wires, vibrating parts, loose elements in vacuum tubes, poor contact in switches, and a countless number of other points all provide their individual contributions.

After stressing the importance of bonding and shielding on an airplane it may seem strange to inject an expression that, in many cases, the practice is overdone. Little or no effort has been expended to determine the extent to which an airplane actually should be shielded. It is true a rather extensive investigation would be required. In designing no one cares to accept the responsibility for interference that might be present. In the course of building and testing there is little time given to delving into such matters. As a result elaborate junction boxes and pull boxes are installed where there is little need for them. Covers are installed over terminal panels with screws and clamps far in excess of any practical requirement. This construction complicates and slows down inspection and maintenance, as well as giving rise to a tendency to pass up that particular point in going over the ship. It is felt that many savings in weight, materials, and installation costs could be made. To sum the matter up in a few words, a ship is shielded largely on suspicion.

Although we are at present enjoying a highly efficient radio operation relatively interference-free, it is obtained by constant vigilance and at a far greater cost than would be necessary if many of the points outlined were given more direct consideration instead of dealing with them in an off-hand manner as it was when radio was first introduced into aviation.